Final Report

Majority Voting Mechanical Feedback Servoactuator for the Saturn S-IV B

Performed Under

NASA Contract NAS8-11634

by

Moog Servocontrols, Inc.
East Aurora, New York

Report Date - July 22, 1965

1.0 INTRODUCTION

Contract NAS8-11634 was awarded to Moog Servocontrols, Inc. on July 6, 1965. The scope of work was defined as follows: "The contractor shall design, develop, fabricate, test and deliver two (2) each servoactuators of the same configuration and meeting the same basic requirements as the Moog Model 17-189 S-IV B servoactuators. These actuators shall employ redundancy techniques to improve reliability."

The following sections describe the successful fulfillment of the contract objectives.

2.0 SUMMARY

The design, development, fabrication, test and delivery of two servoactuators to the basic S-IV B requirements has been completed. These actuators employ redundancy by a technique known as "majority voting." Thru the use of this technique, pertinent output characteristics remain essentially unchanged in consequence to a single failure in any one of the three channels employed. Maximum null offset (due to a failure in a single channel) on the servoactuators built under this contract was less than 4%. This compares very favorably with an initially stated goal of less than 5.0%. The results of testing and evaluating indicate that these limits could be maintained in production thru the use of existing production set-up and test techniques.

Simulator tests indicated adequate performance to the S-IV B requirements. Performance remained adequate with the introduction of simulated failures. Temperature testing to 275°F likewise indicated no adverse performance characteristics.

The observed characteristics of the servoactuators developed under Contract NAS8-11634 are very encouraging with respect to providing redundancy in a servoactuator. In order to evaluate the servoactuators fully, under failure conditions, a larger analysis and test program (than that covered by this contract) is required. Accordingly, an additional contract, NAS8-11998, has been awarded to Moog. This contract covers a reliability analysis, a computer analysis, and an actual induced (not simulated) failure test program.

The following sections describe the actuator design and contain data from tests conducted on the majority voting servoactuators.

3.0 DESIGN DESCRIPTION

3.1 General

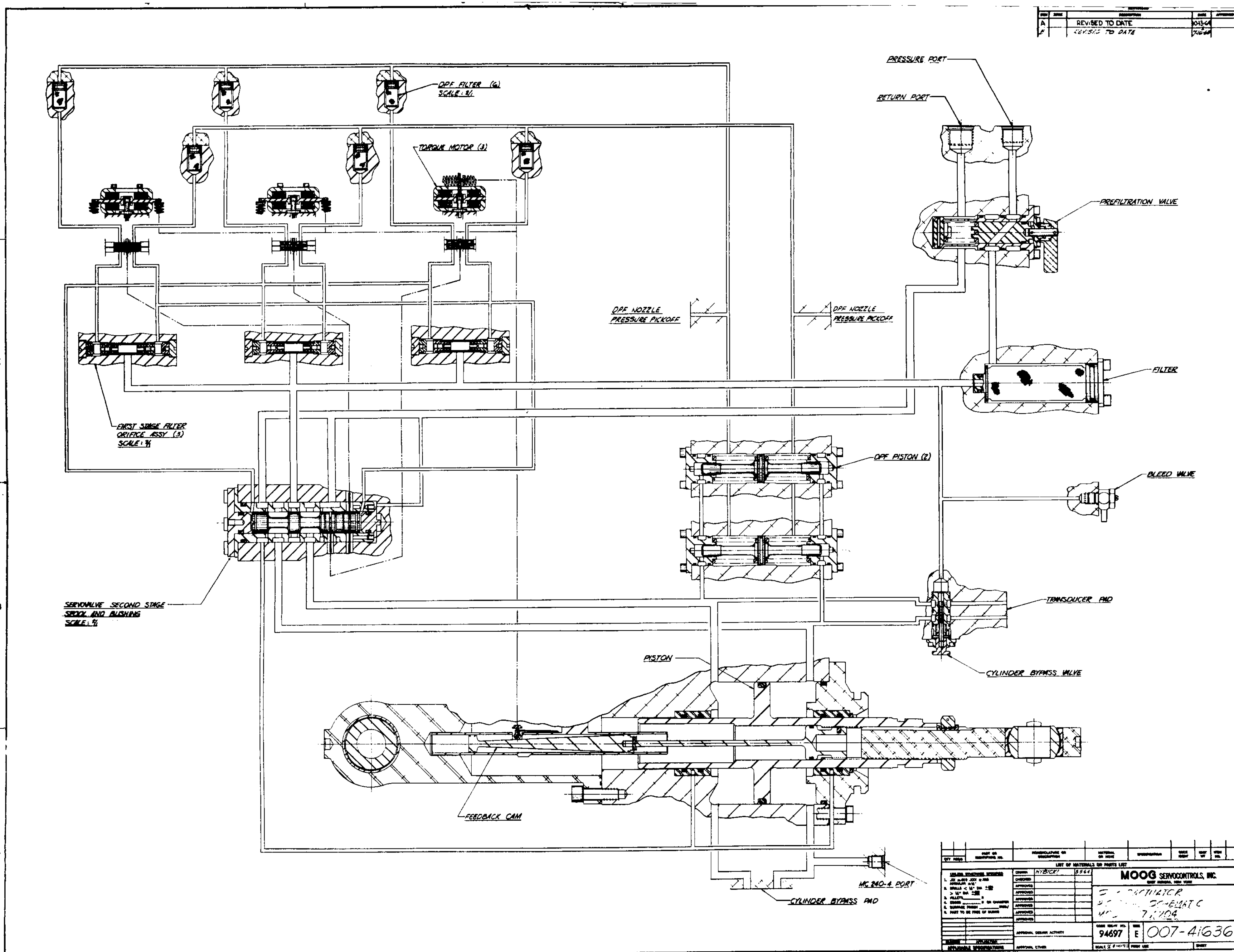
The Model 17-204 majority voting servoactuator produced under this contract is intended for thrust vector control in the Saturn S-IV B stage. The design has the same hydraulic power capacity (piston area and velocity), and physically will attach to the same clevis fittings as the Model 17-189 actuator presently being produced for the S-IV B.

The actuator uses a tapered cylindrical cam for mechanical position feedback and dynamic pressure feedback (DPF) for stabilization of the engine inertia. These features are functionally identical to the current S-IV B actuator. Other similarities to the present S-IV B actuator include: same valve torque motor, same cantilever cam follower assembly, and same accessory components (prefiltration valve, cylinder bypass valve, mid-stroke lock, and vernier scale).

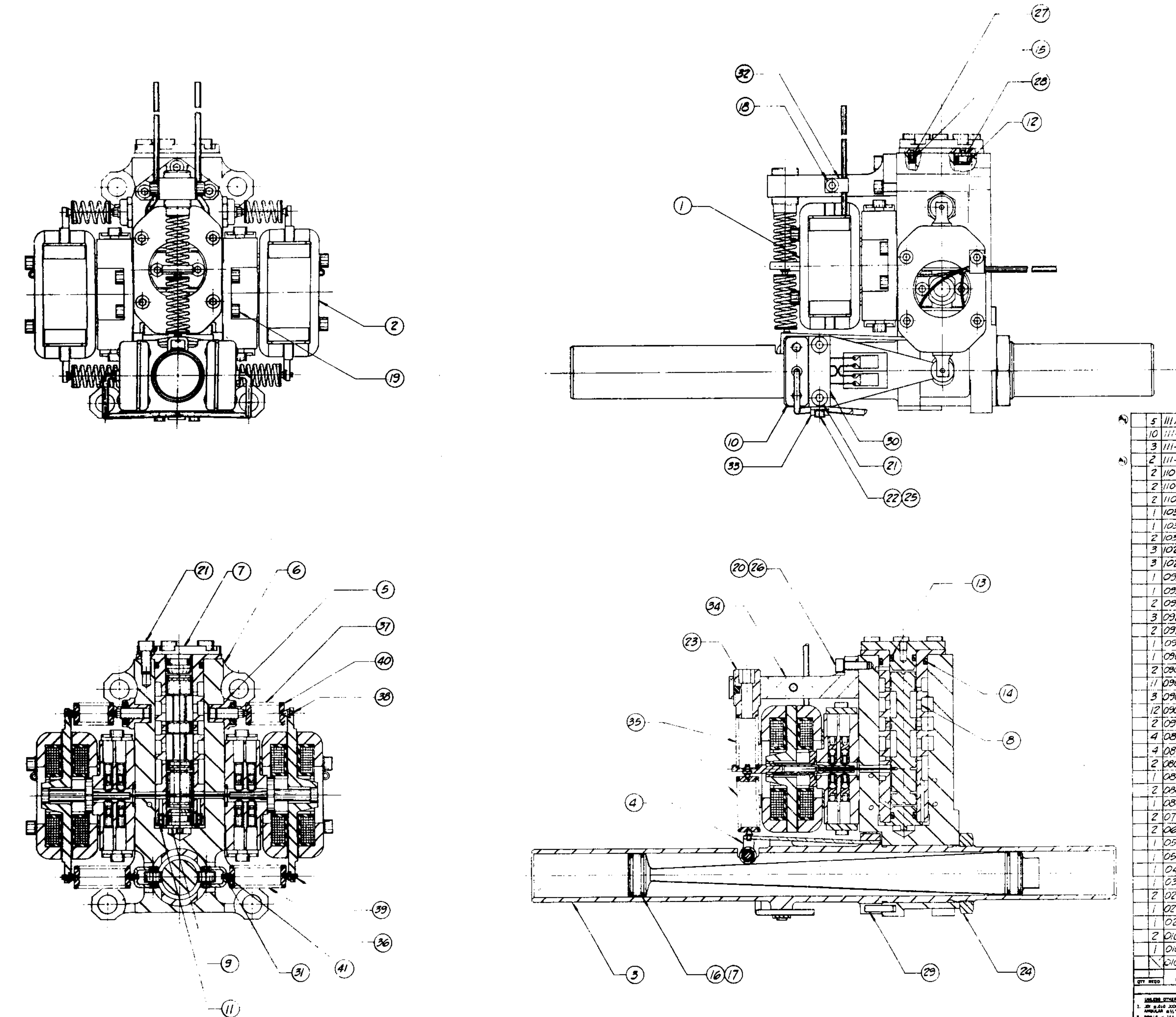
A Pictorial Schematic, 007-41636; Servovalve Assembly Drawing, 010-41833; Servoactuator Assembly Drawing, 010-41933; and Servoactuator Installation Drawing, 001-41932, are included in this report on pages 3-2, 3-3, 3-4, 3-5, and 3-6.

3.2 Operation

The actuator uses three active torque motors in a majority voting configuration. The three torque motors may be driven from a single amplifier, or driven by three separate amplifiers if majority voting is desired for the electrical amplifiers and connecting cables. Each torque motor receives actuator feedback from the cam through separate springs and cam follower assemblies. Each torque motor also receives feedback from the second stage spool. The latter feedbacks are provided by means of cantilever springs attached to each flapper and driven by the single second stage spool of the servovalve. Majority voting is an arrangement whereby the servovalve will behave as the majority of the first stage torque motor hydraulic amplifiers dictate. Since three first stage assemblies are employed, a majority decision will exist when at least two of the first stages


REVISED TO DATE
REVISED TO DATE
PRESSURE PORT
RETURN PORT
PREFILTRATION VALVE
DPF FILTER (6) SCALE: 3/1
TORQUE MOTOR (3)
DPF NOZZLE PRESSURE PICKOFF
DPF NOZZLE PRESSURE PICKOFF
FILTER
FIRST STAGE FILTER ORIFICE ASSY (3) SCALE: 3/1
DPF PISTON (2)
BLEED VALVE
SERVOVALVE SECOND STAGE SPOOL AND BUSHING SCALE: 3/1
TRANSDUCER PAD
PISTON
CYLINDER BYPASS VALVE
FEEDBACK CAM
MC 240-4 PORT
CYLINDER BYPASS PAD
MOOG SERVOCONTROLS, INC.
94697
E
007-41636
007-41636

SYM	ZONE	DESCRIPTION	DATE	APPROVED
A		1. ITEM 38 WAS 7 ... , 111X01459, PIVOT 2. ADDED ITEM 41		



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10	111-41852	SEAT, SPRING					40
3	111-29873	PIVOT, ECCENTRIC					39
2	111-41752	PIVOT					38
2	110-41855	SPRING					37
2	110-41854-2	SPRING					36
2	110-41854-1	SPRING					35
1	103-41850	BRACKET					34
1	103-41845	CLAMP					33
2	103-24927	CLAMP					32
3	102-41853	SHIM					31
3	102-41844	RETAINER, FOLLOWER					30
1	093-41838	DOWEL					29
1	093-41835	PLUG					28
2	093-28184	PLUG					27
3	092-06131-113-187-2	WASHER					26
2	092-06131-093-140-2	WASHER					25
1	091-41839	NUT, RETAINING					24
1	090-41851	ADJUSTOR					23
2	090-22537-45	SCREW, CAP, SCH					22
11	090-06276-100	SCREW, CAP, SCH					21
3	090-06141-11C	SCREW, CAP, SCH					20
12	090-06141-9C	SCREW, CAP, SCH					19
2	090-06141-8C	SCREW, CAP, SCH					18
4	082-41837	CAP, "O" RING					17
4	080-20129-6	"O" RING					16
2	080-04273-38	"O" RING					15
1	080-04273-8	"O" RING					14
2	080-04273-7	"O" RING					13
1	080-04273-3	"O" RING					12
2	071-22762	ORIFICE					11
2	063-41856	POSITION TRANSDUCER ASSY					10
1	052-41772	STOP, SPOOL					9
1	050-41834	BUSHING & SPOOL ASSY					8
1	043-41771	END CAP					7
1	031-41766	BODY					6
2	029-41846	ADJUSTOR ASSY					5
1	029-41842-2	CAM FOLLOWER ASSY					4
1	029-41836	CAM & CAM HOUSING ASSY					3
2	010-41659	FIRST STAGE ASSY MOD 50-278					2
1	010-41694	FIRST STAGE ASSY MOD 50-277					1
	010-41833	SERVOVALVE ASSY					

LIST OF MATERIALS OR PARTS LIST

UNLESS OTHERWISE SPECIFIED

DRAWN NEDESCU

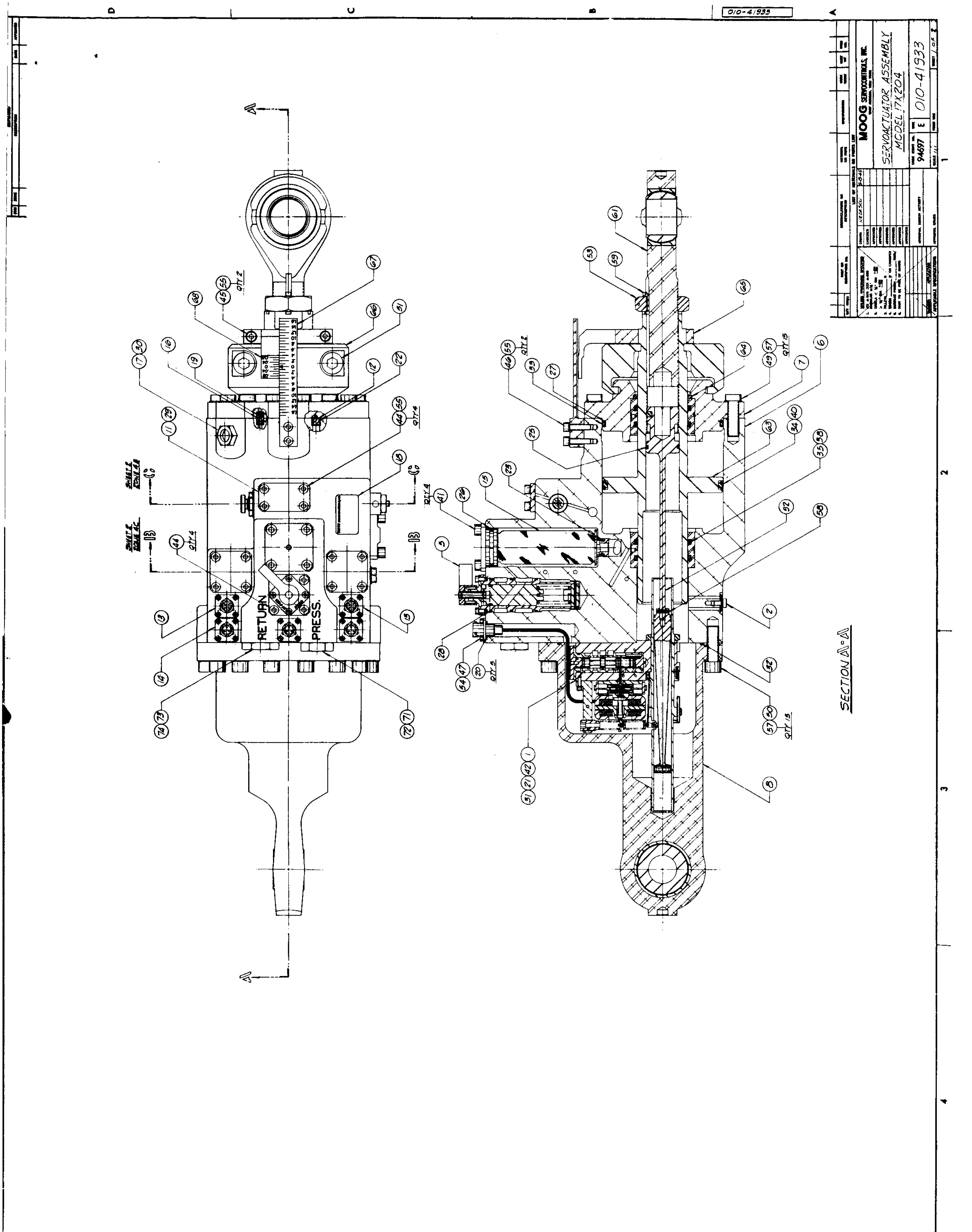
MOOG SERVOCONTROLS, INC. EAST AURORA, NEW YORK

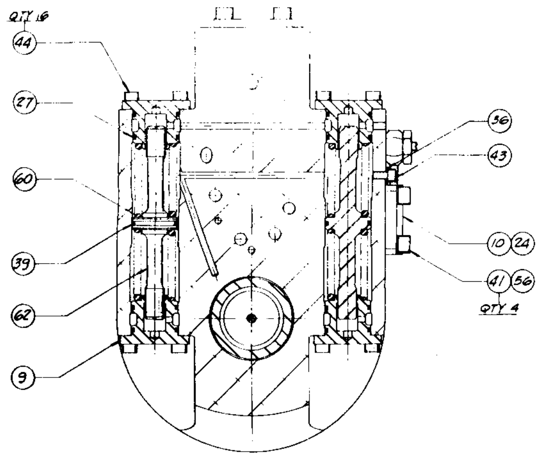
SERVOVALVE ASSEMBLY
MODEL 16-147

CODE IDENT NO.	SIZE	
94697	E	010-41833

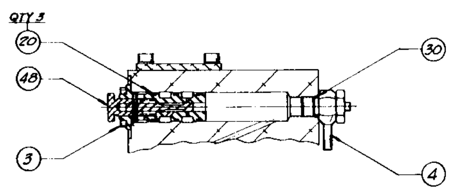
SCALE 2/1

010-41833


010-41933
MOOG SERVOCONTROLS, INC.
SERVOACTUATOR ASSEMBLY
MODEL 17X204
010-41933
SECTION A-A
RETURN
PRESS.



SECTION B-B
ROTATED 90° COUNTER-CLOCKWISE



PARTIAL
SECTION C-C
ROTATED 90° COUNTER-CLOCKWISE

QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATERIAL OR NOTE	SPECIFICATION	CODE IDENT	UNIT WT	ITEM NO.
1	080-20465-0	"O" RING					74
1	080-20465-8	"O" RING					73
1	073-20651-00	PLUG & BLEEDER					72
1	073-20651-80	PLUG & BLEEDER					71
							70
							69
1	131-41942	VERNIER					68
1	131-41939	SCALE					67
1	131-41931	MIDSTROKE LOCK					66
1	151-41150	CLAMP					65
1	131-09084-19	RING, SCRAPER					64
1	130-41928	PISTON, ACTUATOR					63
2	130-29916	PISTON, DPF					62
1	121-29887	BEARING, ROD END					61
4	110-41920-1	SPRING, HELICAL COMPR					60
1	094-20998-7	LOCK, ROD END					59
1	093-20098C3-7	PIN, SPRING					58
30	092-20541CGIGL	WASHER, FLAT					57
4	092-06131-253-375-1	WASHER, FLAT					56
8	092-06131-195-313-1	WASHER, FLAT					55
20	092-06131-113-187-2	WASHER, FLAT					54
1	091-41015	NUT, DRILLED JAM					53
1	090-41938	CAM EXTENSION					52
2	090-29911-59	SCREW, CAP, SCH					51
15	090-29911-46	SCREW, CAP, SCH					50
15	090-29516-620	SCREW, TORQ-SET					49
4	090-06276-10C	SCREW, CAP, SCH					48
20	090-06141-8C	SCREW, CAP, SCH					47
2	090-06132-24S	SCREW, CAP, SCH					46
2	090-06132-18S	SCREW, CAP, SCH					45
24	090-06132-14C	SCREW, CAP, SCH					44
2	090-06130-10C	SCREW, CAP, SCH					43
4	090-06129-24	SCREW, CAP, SCH					42
8	090-06129-20C	SCREW, CAP, SCH					41
1	082-41810-3-4250	CAP-"O" RING					40
2	082-29969-0-1000	CAP-"O" RING					39
4	082-29959-2-1750	CAP-"O" RING					38
1	082-20036-242	RING-BACKUP					37
2	081-44234	DYNASEAL					36
4	080-04273-147	"O" RING					35
1	080-04273-146	"O" RING					34
1	080-04273-139	"O" RING					33
1	080-04273-130	"O" RING					32
2	080-04273-76	"O" RING					31
2	080-04273-69	"O" RING					30
2	080-04273-42	"O" RING					29
3	080-04273-37	"O" RING					28
10	080-04273-38	"O" RING					27
1	080-04273-30	"O" RING					26
1	080-04273-26	"O" RING					25
2	080-04273-22	"O" RING					24
1	080-04273-19	"O" RING					23
1	080-04273-14	"O" RING					22
4	080-04273-13	"O" RING					21
8	080-04273-4	"O" RING					20
2	080-04273-2	"O" RING					19
1	074-29918	PLATE, IDENT					18
1	073-20651-4CL	PLUG & BLEEDER					17
1	075-10970	UNION					16
1	071-29896	FILTER ELEMENT					15
3	061-24826	CONNECTOR, RECP, ELEC					14
2	061-07173	CONNECTOR, RECP, ELEC					13
1	052-22908	SPOOL STOP	USED AS A PLUG				12
1	049-42018	CAP, TRANSDUCER PAD					11
1	049-41851	COVER					10
4	049-29912	END CAP, DPF PISTON					9
1	032-41937	TAILSTOCK & BEARING ASSY					8
1	032-41936	CYLINDER CAP ASSY					7
1	032-41935	ACTUATOR BODY ASSY					6
1	023-426681	PREFILTRATION VALVE ASSY					5
1	023-41525	VALVE, BLEED & SAMPLING					4
1	023-29976	CYL BYPASS VALVE ASSY					3
1	023-12275	VENT, CHECK					2
1	010-41833	VALVE ASSY (16-147)					1
X	010-41933	SERVOACTUATOR ASSY					

LIST OF MATERIALS OR PARTS LIST

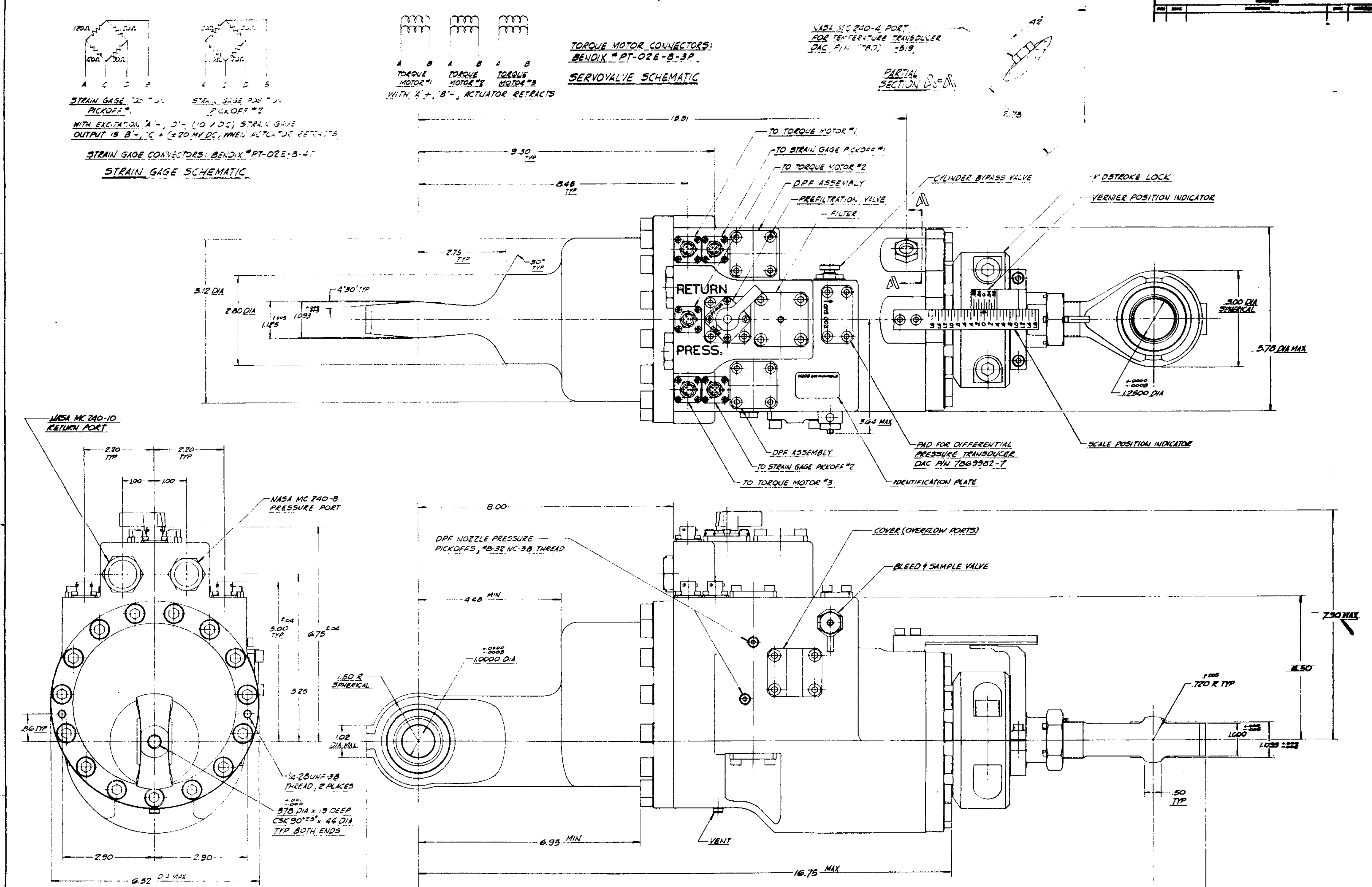
MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

SERVOACTUATOR ASSEMBLY
MODEL 17X204

CODE IDENT NO. 94697 | SIZE E | 010-41933

SCALE 1/1 | SHEET 2 OF 2

010-41933



NOTES:
1. PART NUMBER: 010-41933
2. ACTUATOR EXTERIOR: STAINLESS STEEL, ALUMINUM ALLOY (ANODIZED), TITANIUM, CADMIUM PLATE & ENAMEL PER MIL-E-7729).
3. FLUID: MIL-H-5606, 5 MICRON NOMINAL FILTRATION.
4. OPERATING TEMPERATURE RANGE: -65°F TO +275°F.
5. SYSTEM PRESSURE: 3650 PSI
6. PISTON AREA: 11.78 SQUARE INCHES, STROKE ± 7.00 INCHES
7. DRY WEIGHT: 46 LBS
8. ASSEMBLY DATE PER ANA 438 STAMPED NEAR IDENT. PLATE.
9. EXTERNAL O RING COMPOUND: BUNA N PER MIL-P-25732.

MOOG SERVOCONTROLS, INC.

SERVOACTUATOR INSTALLATION

MODEL 17X204

94697 E 001-41932

are in agreement.

Majority operation is possible because of the ability of the armature of the torque motor to be essentially a free beam sensitive to force balances. For instance, if one torque motor should receive a hardover command, then its torque motor would be satisfied only when a hardover feedback to the torque motor existed. However, the remaining two torque motors would be satisfied when the feedback torque equalled their input torque, which for this example we can say is zero (actuator centered). The action that would take place for this example is as follows: (1) The flapper of the hardover torque motor would seal against one nozzle of its hydraulic amplifier, (2) the flappers of the remaining two torque motors would seek a position such that equal hydraulic pressures would continue to exist at the second stage spool end chambers, (3) the servovalve would remain at null and the actuator centered. The hardover torque of the failed motor would be balanced by the mechanical torque produced at the sealed nozzle. The zero torque input of the remaining two (or majority of the) motors would be satisfied by the actuator centered position. Similar analogies can be made for the cases of nozzle plugging, fail neutral, plugged inlet orifice, and partial failure of any one first stage torque motor hydraulic amplifier assembly. The majority voting servovalve will also vote if a failure occurs in one of the connecting cables of the servoamplifiers.

The method of providing DPF consists of an auxiliary pair of nozzles to each torque motor, together with two spring centered pistons. Load pressures are applied to either end of the nozzle on one side of each torque motor flapper. The nozzle-flapper acts as a resistance that is sensitive to flow; that is, its resistance increases with increased flow. The spring centered pistons act as capacitances which are coupled with the nozzle-flappers to form a frequency sensitive network.

Static or very low frequency load pressure changes result in displacement of the auxiliary pistons against their springs. As the frequency of load pressure change is increased, the fluid of the annular sections of the pistons must attempt to pass by the nozzle-flappers at ever increasing rates. Because the nozzle-flapper resistance is sensitive to flow, at higher frequencies the resistance increases and pressures proportional to load pressures (by the area ratio of the auxiliary pistons) are imposed on the flapper. These dynamic load pressures produce a torque on each flapper. Servovalve performance is thus modulated by this action and feedback polarity is arranged so as to produce damping. As was the case for basic servovalve operation, three sets of auxiliary nozzles are utilized in order to retain majority voting capability. The use of two spring centered pistons provides redundancy in that if one piston fails (seizure or broken spring), the DPF feature will be retained. In this case, the frequency at which load damping becomes effective will increase. Sizing is such that damping takes place well below the resonant frequency.

Additional auxiliary features are provided in the servoactuator. Items included are a prefiltration valve, a cylinder bypass valve (manual and hydraulically operated) and a strain gage telemetering device. Ground support items include a removable mid-stroke lock and a position indicator scale and vernier.

3.3 Design Features

The actuator design is based upon the present 17-189 (S-IV B) actuator. An aluminum alloy barrel is used with a high expansion steel alloy piston. In order to maintain the present S-IV B end fitting, it was necessary to reduce the stroke of the actuator from approximately ±1 1/2 inches to ±1 1/4 inches. The reduction in stroke was caused by the valve configuration. The majority voting servovalve maintained the rugged straightforward design of the present S-IV B valve. The basic fundamental that was followed in the servovalve design was to not compromise reliability in order to package added components. Just as in the S-IV B servovalve, there are no levers, links or gimmicks used in the feedback mechanism. However, a price must be paid for this type of straightforward design, and that price is added actuator length or reduced actuator stroke. Since the actuator length was fixed and there is good reason to assume that the actuator stroke could be reduced, the latter course was taken.

The valve is a modular design. It consists of three separable first stage torque motor assemblies, a second stage spool-bushing assembly, and a cam, cam housing assembly mounted to a stainless steel valve body. The first stage torque motor assemblies have been made as identical to one another as possible, to minimize the differences in thermal, dynamic, and vibrational characteristics. Two of the first stage assemblies utilize the balanced armature feedback concept of the S-IV B valve; the third first stage utilizes the extended flapper feedback necessitated by the design limitations discussed in the previous paragraph.

The first stage torque motor assemblies contain the torque motor, flapper and first stage nozzles, first stage filter orifice assembly, DPF nozzles, and individual DPF nozzle filters.

The second stage spool bushing design is straightforward. The bushing is flange mounted and has a laminar fit into a dead ended bore. The flange mounted bushing is employed to compactly package the cam, cam housing assembly at the opposite end of the valve body. The amplifier passages from the first stage nozzles terminate in common chambers at the ends of the spool. The DPF nozzles are joined in a common annular bushing groove and ported out through the base of the valve to the DPF pistons.

The cam and cam housing mount into a lap fit hole in the valve body. The cam housing is restrained from rotational motion by a keyway and from translational motion by a tie-wired nut. The three leaf spring followers attach to the cam housing. These followers are the same as those used on the S-IV B valve. However, the leaf spring used on the flapper extension feedback assembly is made of a non-magnetic material because it operates in close proximity to the air gap of the torque motor.

The DPF pistons for the servovalve are located in the actuator. This is the same arrangement used on the S-II and the S-IV B actuators. Two DPF pistons are used to provide redundancy. Only two pistons are required to provide redundant operation because their mode of failure is passive.

Redundant connectors are provided for the servovalve in order to maintain the majority voting concept all the way to the servo-actuator input.

Telemetry information on actuator position is accomplished by strain gage elements mounted on two of the cantilever beam cam follower arms. This method of obtaining actuator piston information has been developed by Moog to eliminate the telemetry potentiometer. Tests conducted on these devices indicate that they are as good or better than potentiometers and they add no extra weight. More importantly, the use of the strain gage elements eliminates the reliability compromise of driving the feedback cam with the shaft of the telemetry pot.

The prefiltration valve and the cylinder bypass valve are of a modular design identical to those now being used in the S-IV B. The actuator drawing indicates provisions for mounting the same customer supplied equipment; i.e., temperature transducer and differential pressure transducer, as the S-IV B.

The mid-position lock and the position indicator are similar in design to those now used on the S-IV B. They have been modified to take advantage of the shorter actuator stroke.

The actuator materials are stainless steel, anodized aluminum, cadmium plated steel, and titanium. Their usage is the same as in the present S-IV B actuator.

The weight of the majority voting actuator is 46 pounds compared to the present S-IV B actuator weight of 40 pounds.

The design of the actuator is characterized by rugged straightforward construction. The ultimate goal was the highest possible unit reliability. Where compromises had to be made they were made at the expense of stroke, weight or envelope.

4.0 TEST RESULTS

Assembly of the servoactuator presented no gross problems. Minor mechanical type problems requiring some dimensional changes to detail parts were encountered, but this is typical of first hardware in any development program. During set-up of the servovalve, magnetic interaction of the torque motors due to their proximity to each other was observed and caused some problem. Recognition of the problem led to suitable set-up techniques which negated the interaction.

Views of a completed actuator are given in Fig. 1 and 2. An exploded view is given in Fig. 3. The servovalve assembly and partial views of the servovalve are given in Figures 4, 5, and 6.

The servoactuators successfully met all performance goals. These goals were to perform within the S-IV B requirements, with and without failures in a single channel, and to exhibit less than 5% shift in actuator position following a failure.

The expected performance was indicated by Moog Report ER-88. This report indicated expected performance at a position loop gain, K_{vx}, of 20 sec^{-1}, and a dynamic pressure loop gain, K_{vp}, of 33.2 sec^{-1} and presented the sizing calculations used in establishing the gain values. The servovalves for the program were initially built to the values specified in ER-88. However, as a result of the development testing of the S-IV B system, it became apparent that system resonance was at a lower frequency than that originally calculated. Original system resonance had been stated as 57.1 rad/sec, and the later value was 43.0 rad/sec. Production S-IV B actuator gains were adjusted in order to provide acceptable system performance. Discussions between NASA (M. Kalange) and Moog Engineering (W. J. Thayer) in November of 1964 resulted in agreement to change the S-IV B majority voting actuator to have identical gains to that of the production S-IV B. Revision ER-88A reflected the necessary sizing changes and indicated expected performance. Moog reports ER-88 and ER-88A are included in the appendix of this report. A nominal position loop gain, K_{vx}, of 14.25 sec^{-1} and a dynamic pressure loop gain of 20.4 sec^{-1} is indicated in ER-88A. The actuators built under this program had gain values as follows:

	K_{vx}	K_{vp}
Actuator S/N 1	16.1 sec^{-1}	17.6 sec^{-1}
Actuator S/N 2	15.8 sec^{-1}	20.2 sec^{-1}

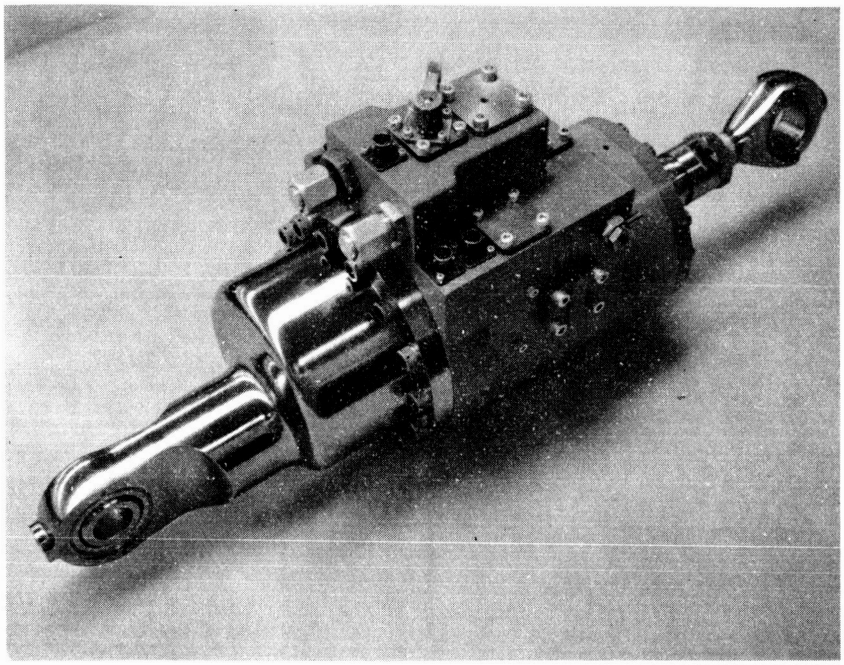


Fig. 1 Completed S-IV B Majority Voting Actuator

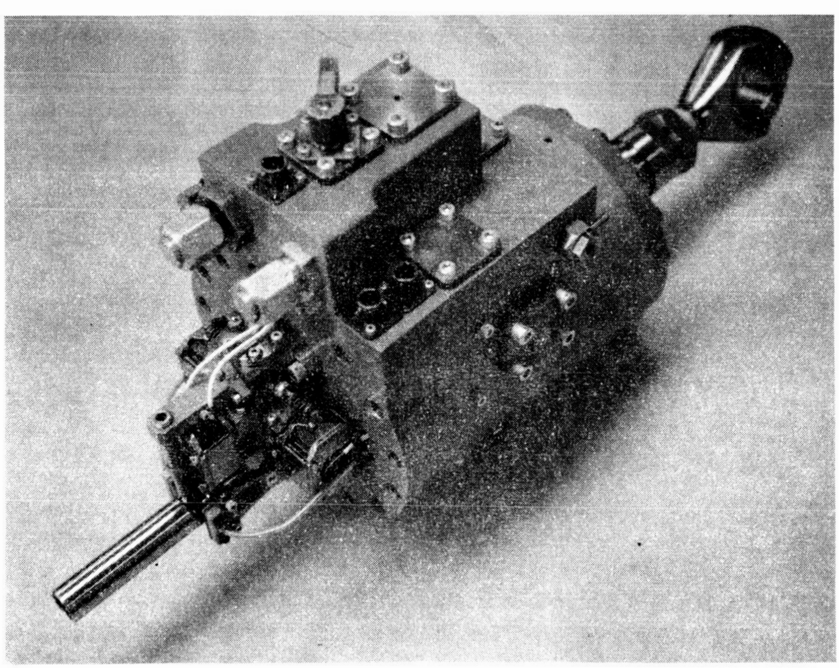


Fig. 2 Completed S-IV B Majority Voting Actuator with Tailstock Removed

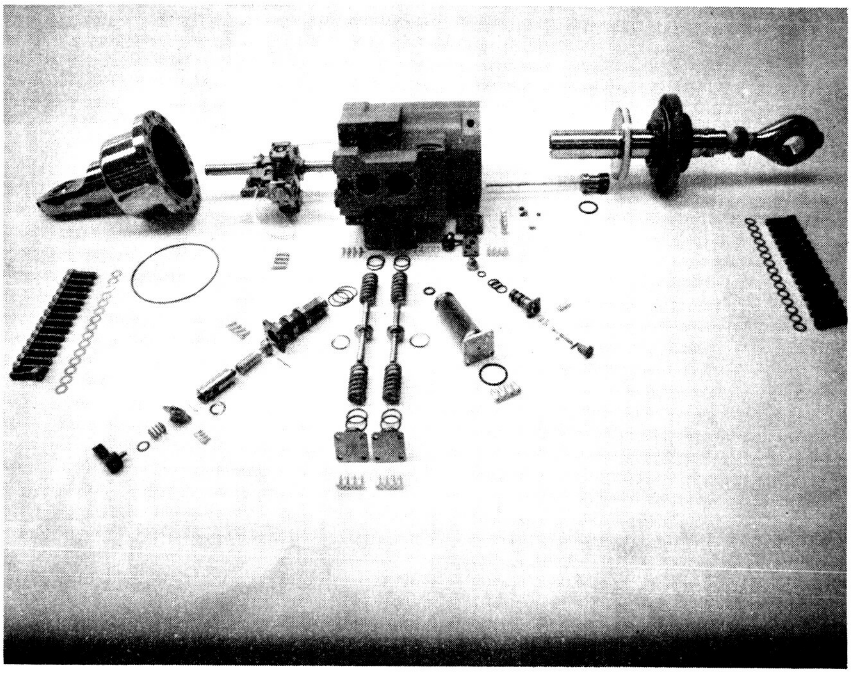


Fig. 3 Exploded View of S-IV B Majority Voting Actuator

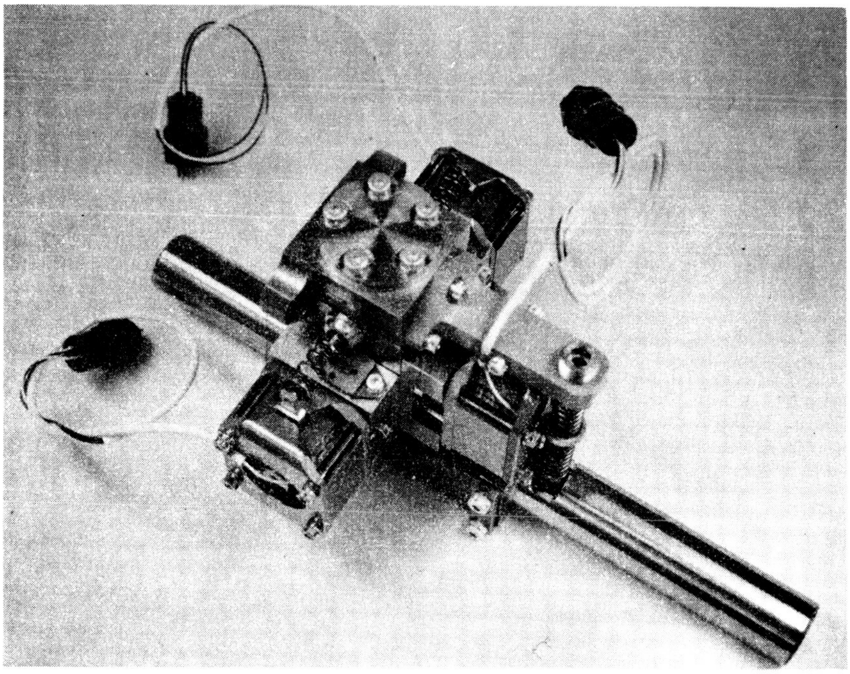


Fig. 4 Majority Voting Valve, Cam Housing, and Cam Assembly

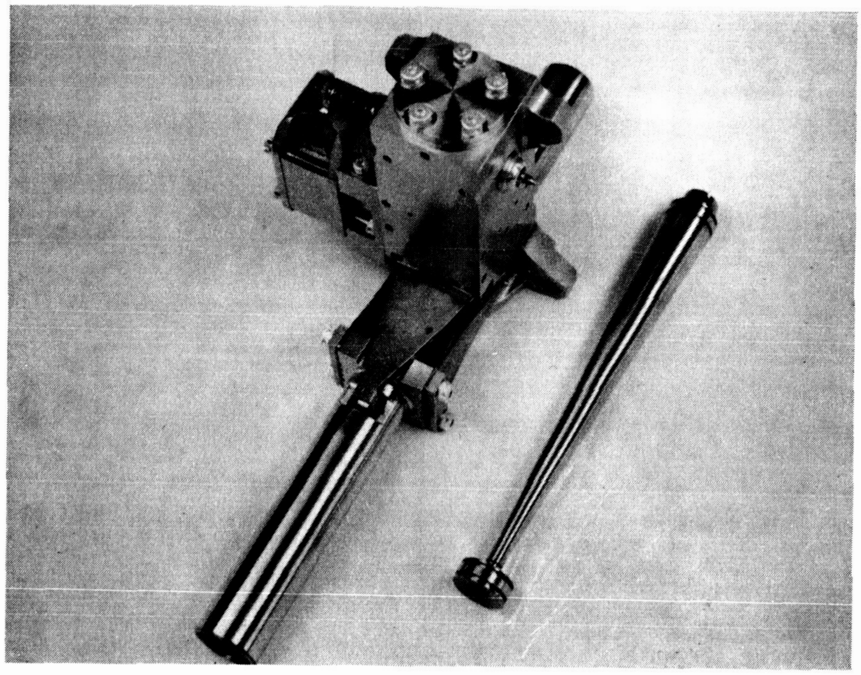


Fig. 5 Partial Assembly: Valve, Cam Housing, and Cam

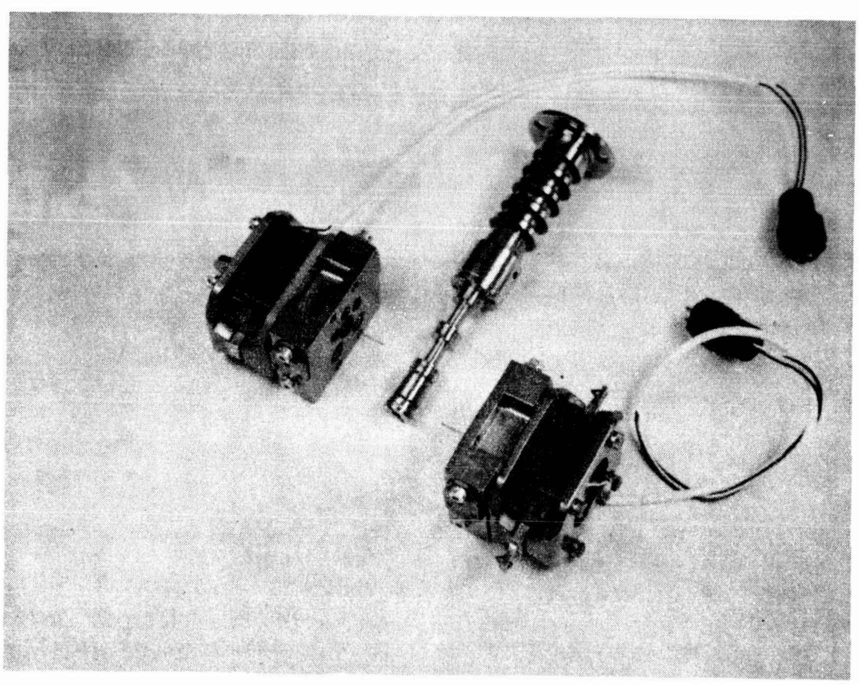


Fig. 6 Detail Parts: Torque Motor Assemblies and Valve Spool - Bushing Assembly

The following pages contain data typical of the two actuators. All data is from Actuator S/N 2.

Figure 7 is a plot of actuator position vs. input signal with all channels active. Relationships such as midstroke lock to extend and retract positions, and mid-stroke lock to telemetry strain gage zero are also given on this plot. Note that the indicated position (as given by strain gage output) and actual actuator position differ by one and one-half percent from mid-stroke to either full extend or full retract due to the non-linearity of the strain gage output. Figures 8, 9, and 10 indicate the output characteristics of the actuator with open and hard-over failures in each of the three channels. Figures 11, 12, and 13 are plots taken under the same conditions, but having each plot run with its own axis rather than a common axis in order that a more accurate assessment can be made of actuator change in position following the introduction of a simulated failure. Maximum shift observed was 4% (see Fig. 13).

Dynamic characteristics of the servoactuator with and without simulated failures were also assessed. The servovalve's open loop dynamic characteristics under conditions of no failure, hard-over failure of one torque motor, or open (coil) failure of one torque motor are given in Figures 14 thru 19. These figures indicate dynamic characteristics for the case of failure of the three motors, one at a time, and also reflect the effect of input amplitude. Figures 20 and 21 are plots of dynamic characteristics taken on the S-IV B load simulator at two different command amplitudes to the servoactuator. From the plots it can be seen that the dynamic characteristics remain essentially unchanged following an induced failure in one channel. Comparison of the actual data with the expected performance predicted in ER-88A indicates that phase lag is less than predicted and the amplitude ratio has a higher peak value than predicted. A portion of this discrepancy can be attributed to the fact that the actual loop gain of the servoactuator was somewhat higher than nominal gain (15.8 sec^{-1} vs. 14.25 sec^{-1}), but it is also true that greater peak amplitude is generally seen on actual test due to mechanical characteristics of the actuator and load simulator, such as back-lash of bearings.

Step input tests at various command inputs and with simulated failures were also conducted on the load simulator. Observed characteristics are illustrated by the photographs of Figures 22 thru 27. Again, it can be observed that transient response remains essentially unchanged following an induced failure in one channel.

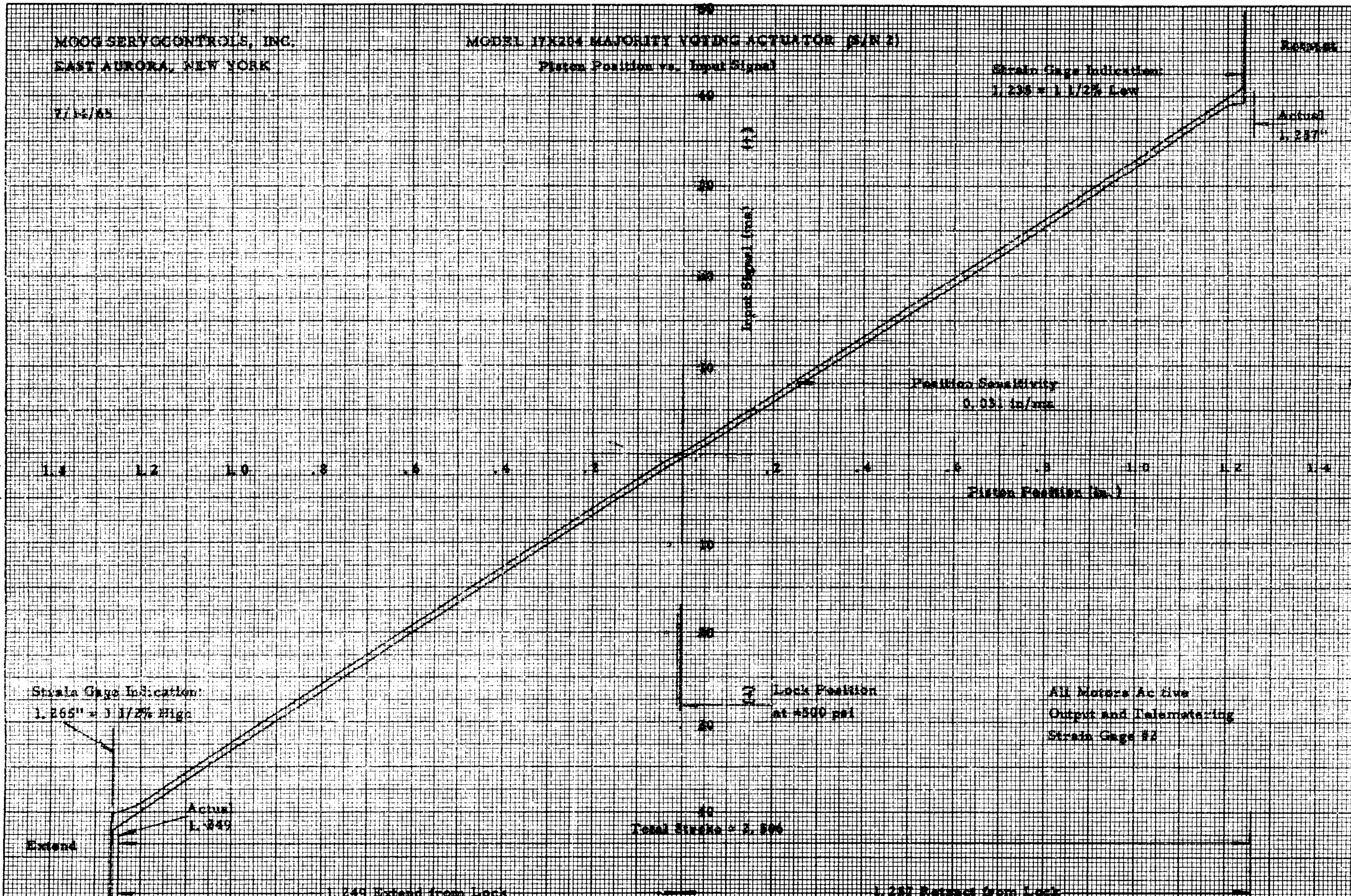
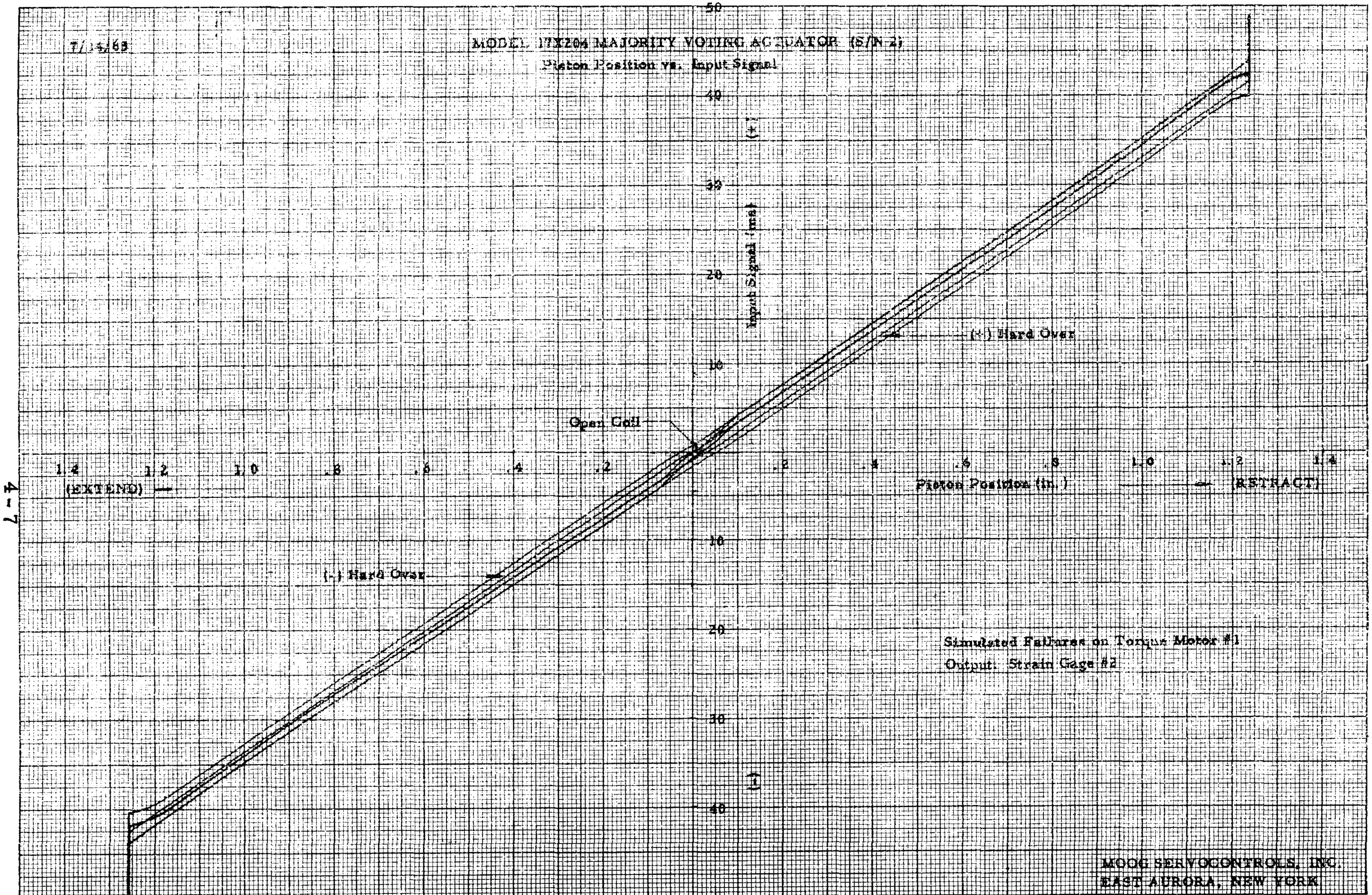

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK
7/14/65
MODEL 17X204 MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
Retract
Strain Gage Indication:
1.238 = 1 1/2% Low
Actual
1.257"
Input Signal (ma)
(+)
Position Sensitivity
0.031 in/ma
Piston Position (in.)
1.4
1.2
1.0
.8
.6
.4
.2
50
40
30
20
10
(-)
Lock Position
at 4500 psi
All Motors Active
Output and Telemetering
Strain Gage #2
Strain Gage Indication:
1.265" = 1 1/2% High
Actual
1.249
Extend
Total Stroke = 2.506
1.249 Extend from Lock
1.257 Retract from Lock

Fig. 7



7/14/65
MODEL 17X204 MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
50
40
30
20
10
10
20
30
40
50
(+)
Input Signal (ma)
(-)
(+) Hard Over
Open Coil
(-) Hard Over
1.4
1.2
1.0
.8
.6
.4
.2
.2
.4
.6
.8
1.0
1.2
1.4
(EXTEND)
Piston Position (in.)
(RETRACT)
Simulated Failures on Torque Motor #1
Output: Strain Gage #2
MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 8

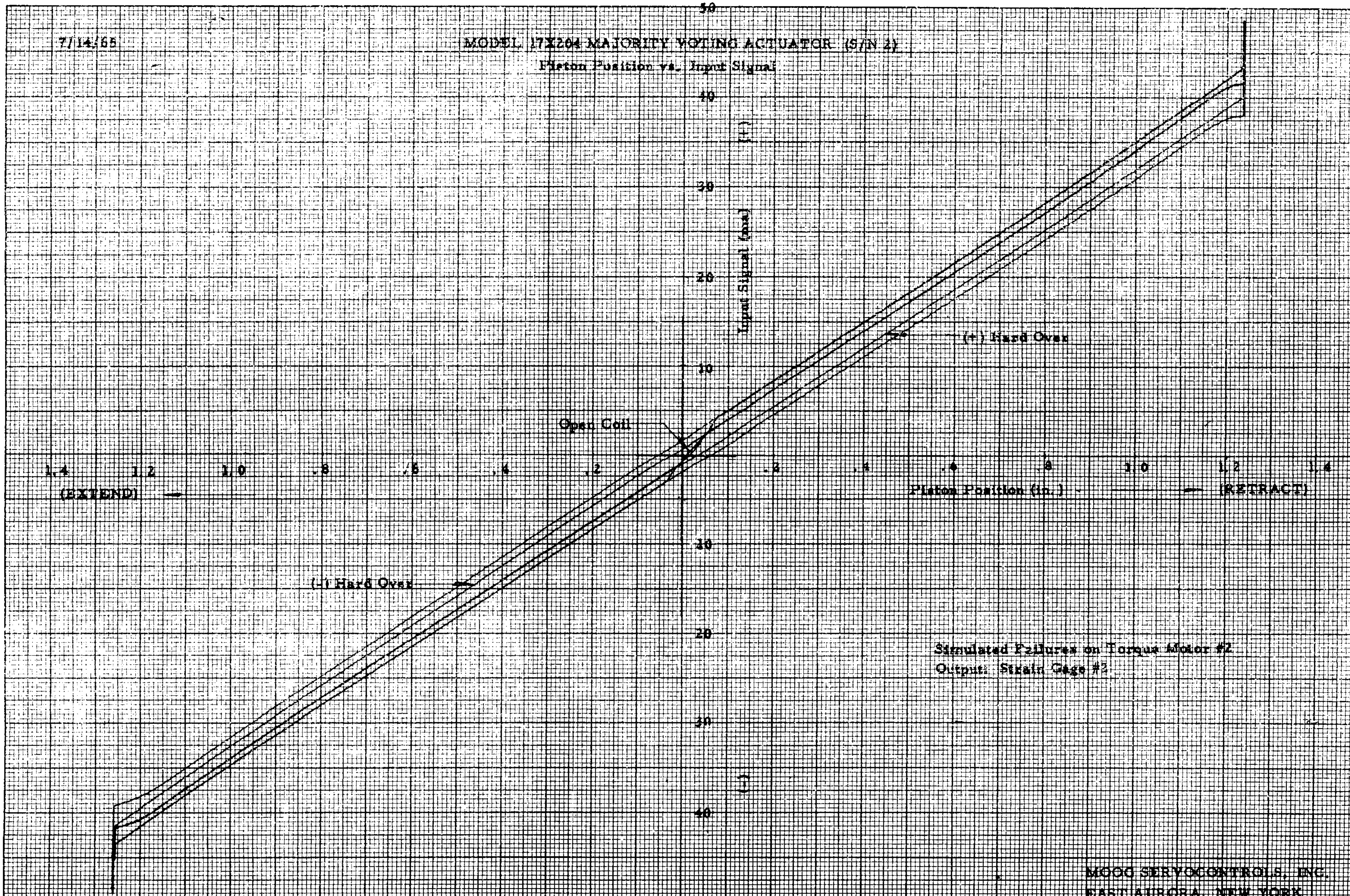

7/14/65
MODEL 17X204 MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
Input Signal (ma)
(+)
(-)
Open Coil
(+) Hard Over
(-) Hard Over
(EXTEND)
(RETRACT)
Piston Position (in.)
Simulated Failures on Torque Motor #2
Output: Strain Gage #2
MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 9

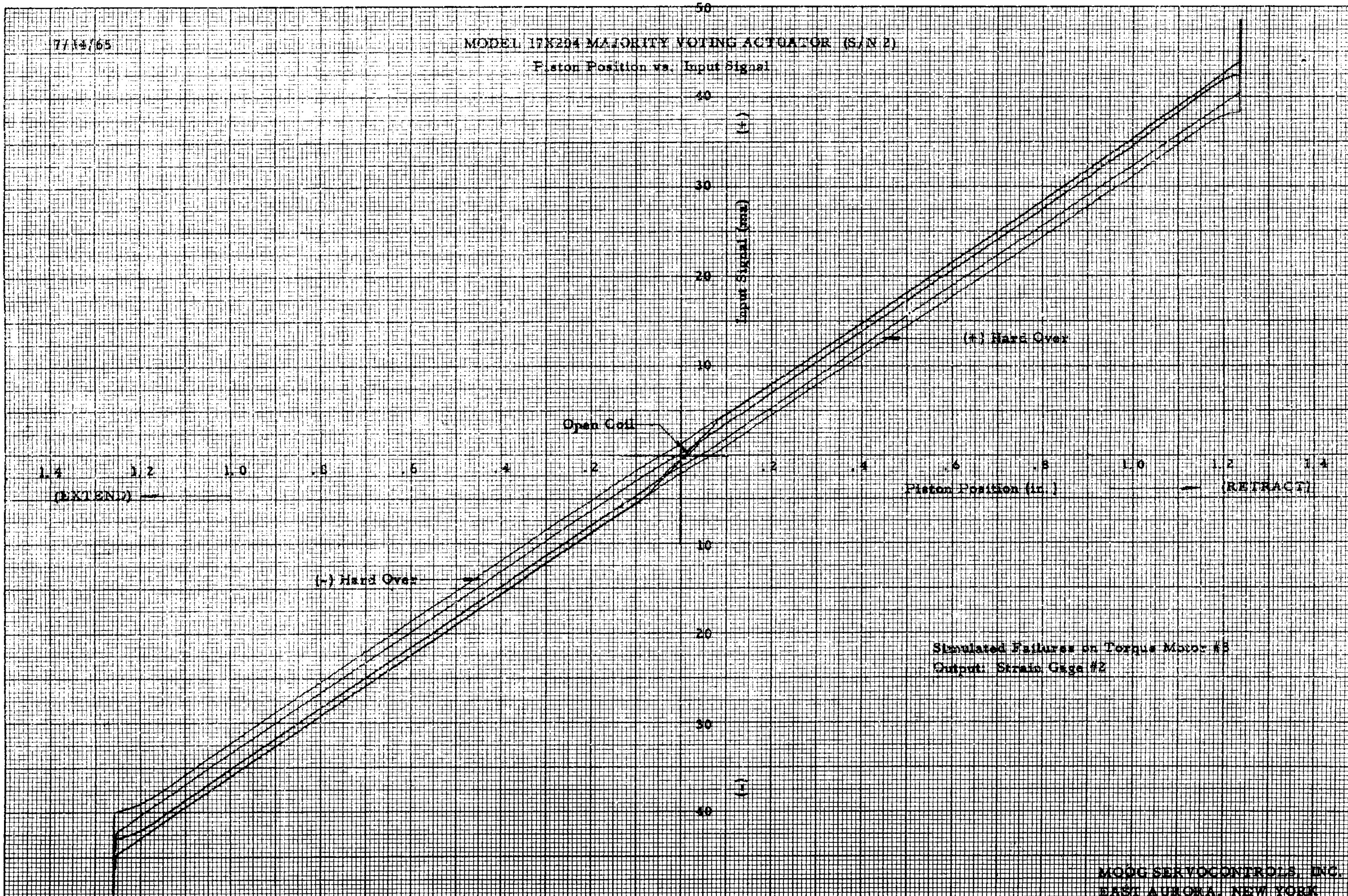

7/14/65
MODEL 17X204 MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
Input Signal (ma)
(+)
(-)
Piston Position (in.)
(EXTEND)
(RETRACT)
Open Coil
(+) Hard Over
(-) Hard Over
Simulated Failures on Torque Motor #3
Output: Strain Gage #2
MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK
50
40
30
20
10
10
20
30
40
50
1.4
1.2
1.0
.8
.6
.4
.2
.2
.4
.6
.8
1.0
1.2
1.4

Fig. 10

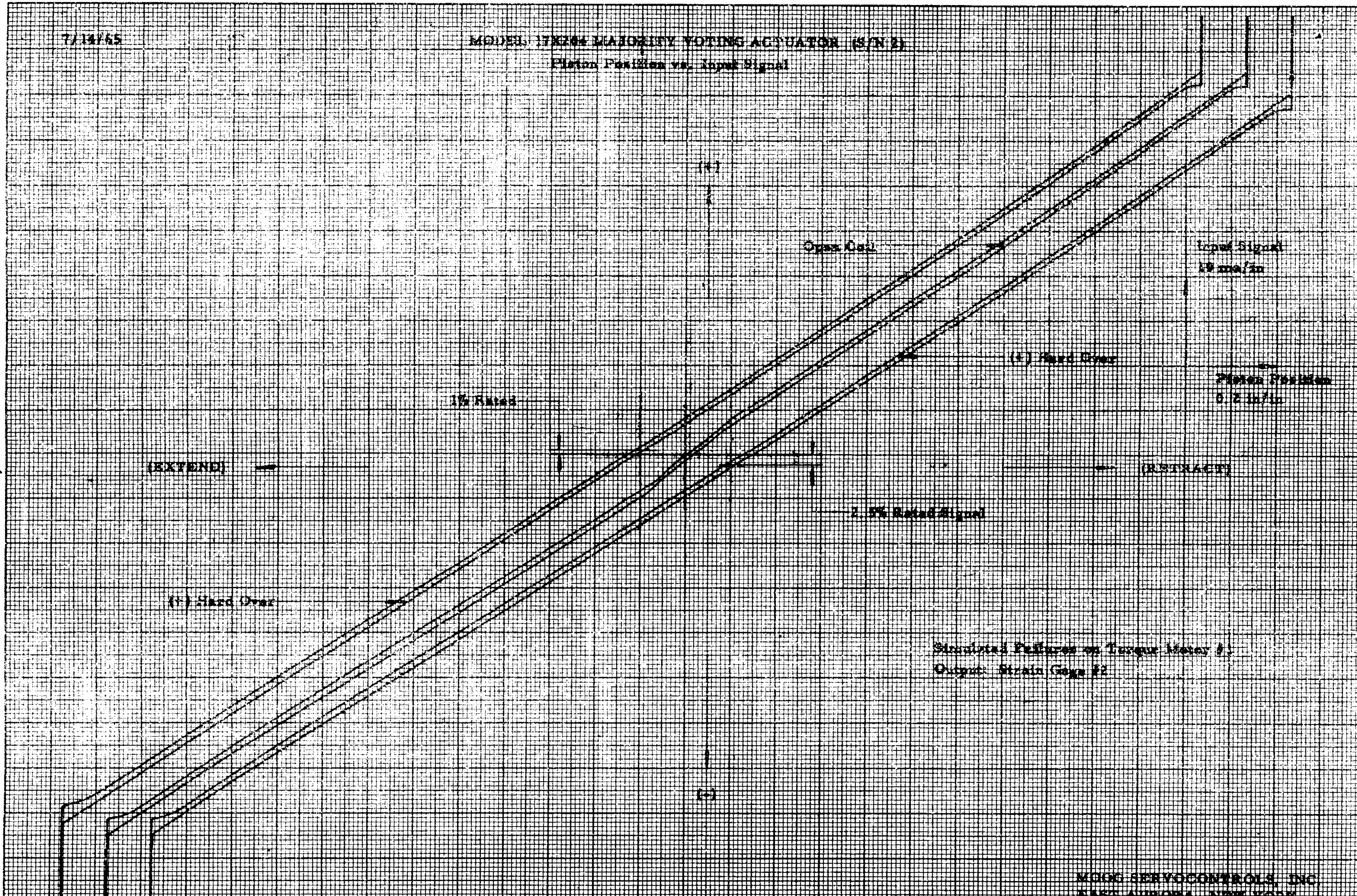
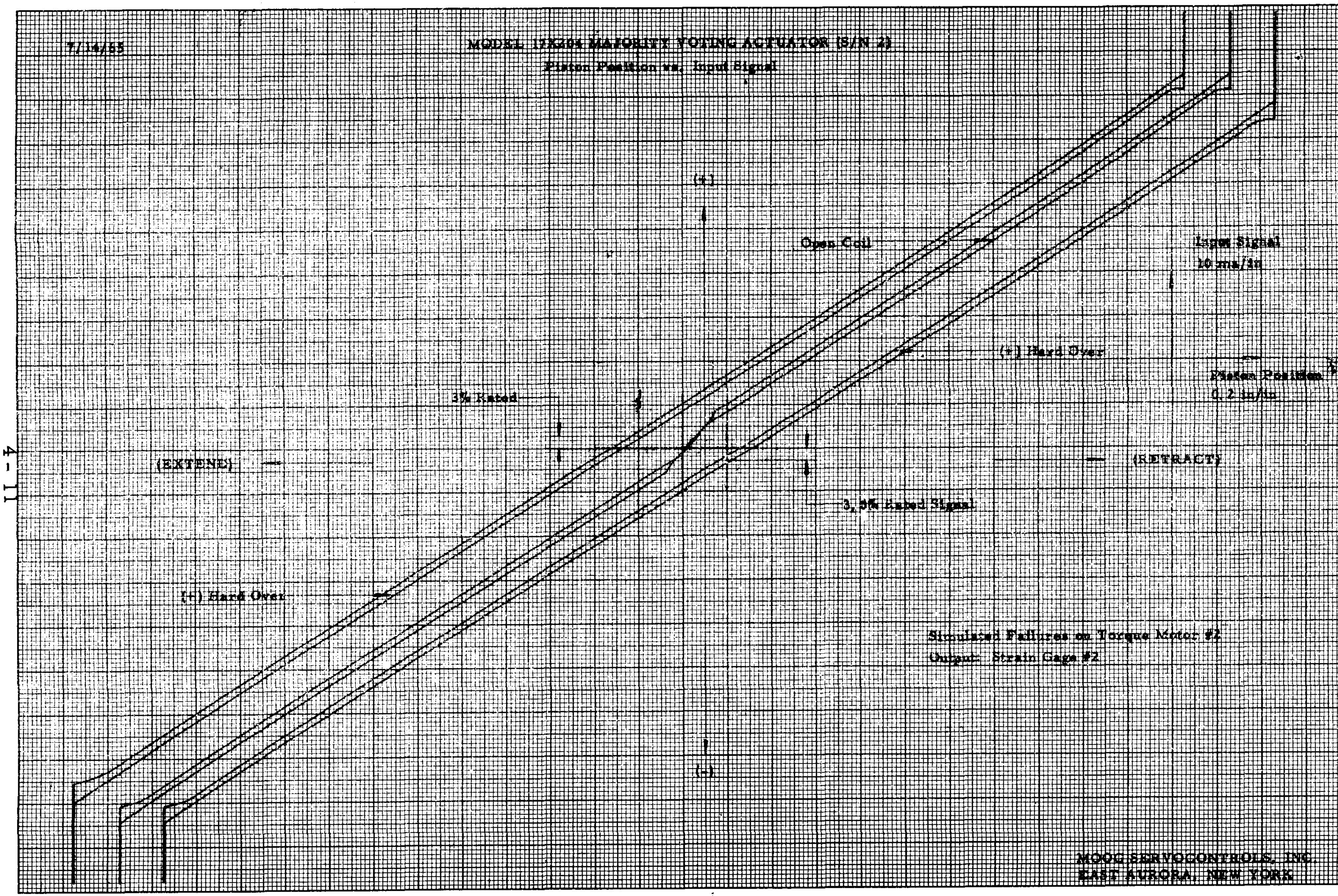

7/14/65
MODEL 17X264 MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
(+)
Open Coil
Input Signal
19 ma/in
(+) Hard Over
Piston Position
0.2 in/in
1% Rated
(EXTEND)
(RETRACT)
2.5% Rated Signal
(-) Hard Over
Simulated Failures on Torque Motor #1
Output: Strain Gage #2
(-)
MOOG SERVOCONTROLS, INC
EAST AURORA, NEW YORK

Fig. 11



7/14/65
MODEL 17X204 MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
(+)
Open Coil
Input Signal
10 ma/in
(+) Hard Over
Piston Position
0.2 in/in
3% Rated
(EXTEND)
(RETRACT)
3.5% Rated Signal
(+) Hard Over
Simulated Failures on Torque Motor #2
Output: Strain Gage #2
(-)
MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 12

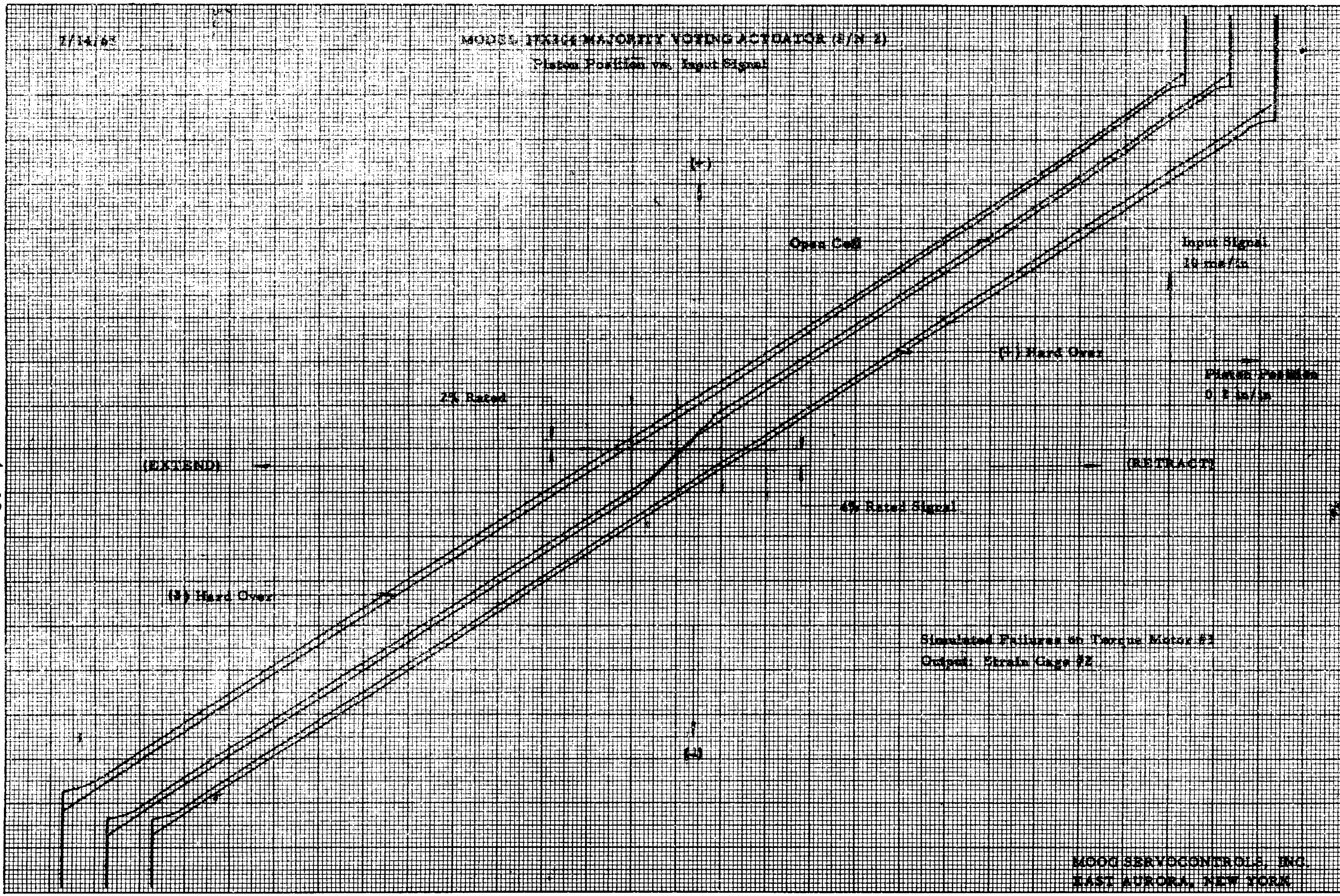
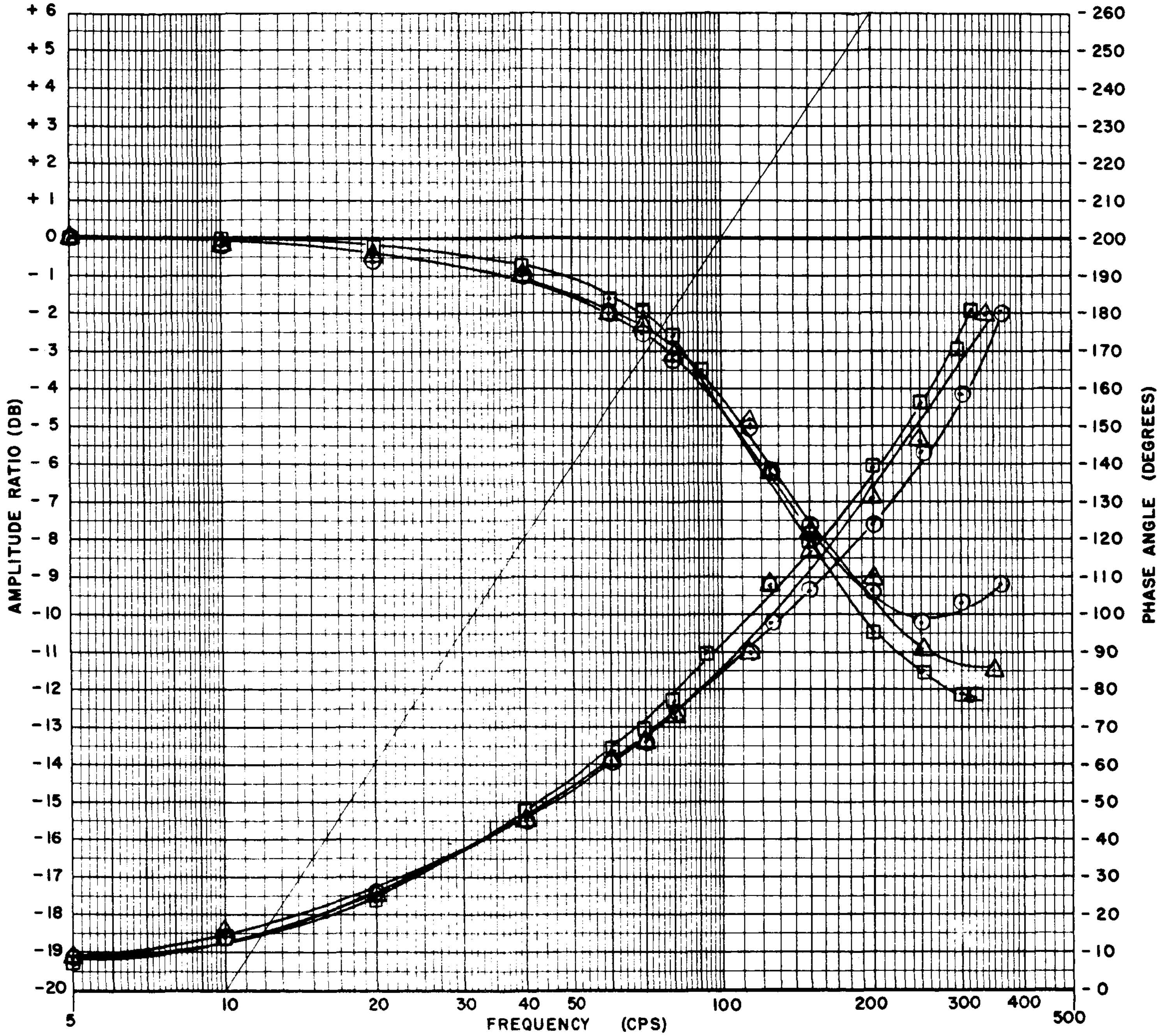

MAJORITY VOTING ACTUATOR (S/N 2)
Piston Position vs. Input Signal
(+)
(-)
Open Coil
Input Signal
10 ma/in
(+) Hard Over
Piston Position
0.2 in/in
2% Rated
(EXTEND)
(RETRACT)
4% Rated Signal
(-) Hard Over
Simulated Failures on Torque Motor #1
Output: Strain Gage #2
MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 13

SERVO VALVE DYNAMIC RESPONSE



MODEL 16-147 SERIAL 3

DATE 6-2-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 1 MA.(P-P)

OUTPUT No Load Flow

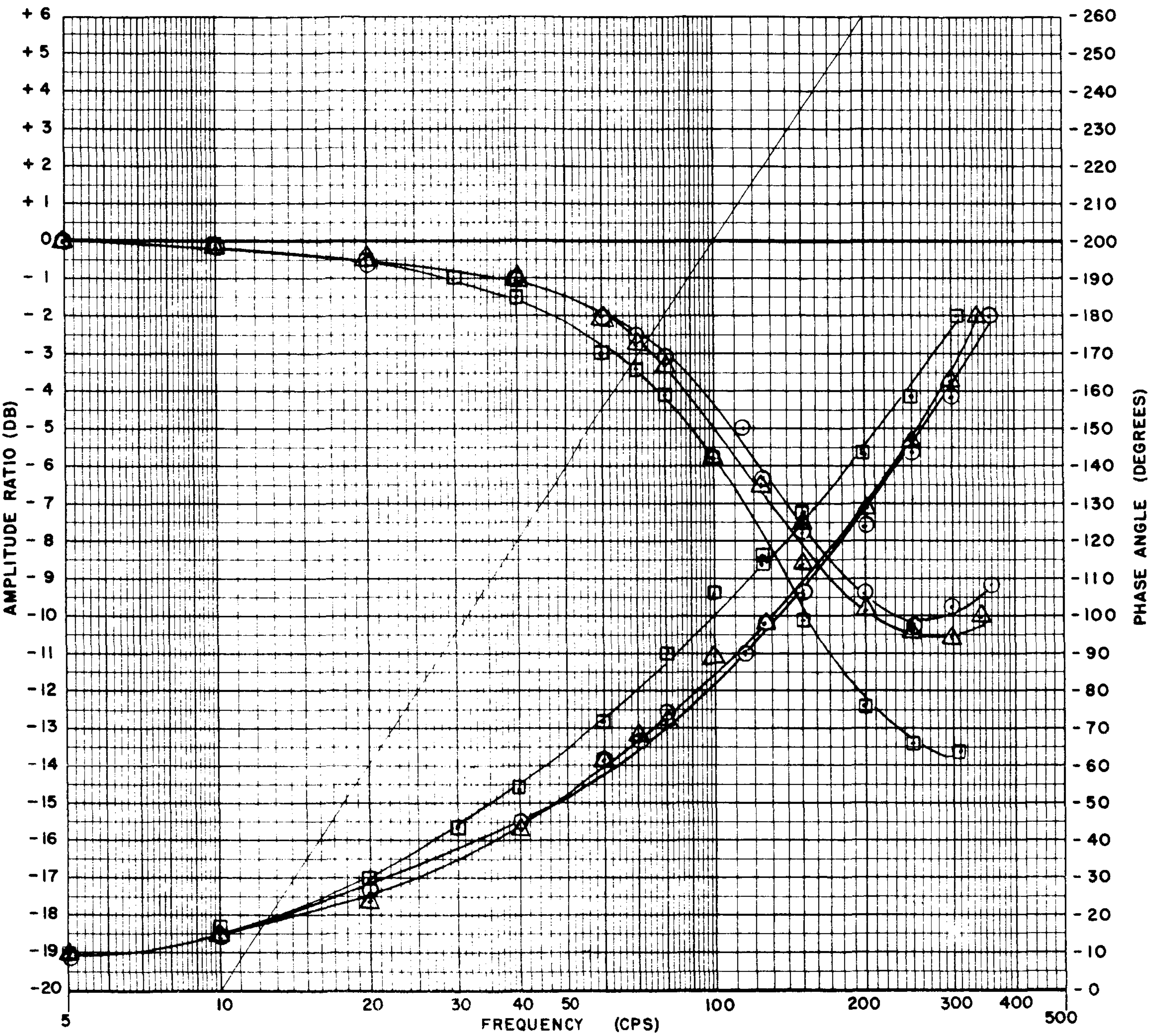
Torque Motor #1 Failed

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 14

○ - No Failure
△ - Open
□ - +50 ma

SERVO VALVE DYNAMIC RESPONSE



MODEL 16-147 SERIAL 3

DATE 6-2-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 1 MA.(P-P)

OUTPUT No Load Flow

Torque Motor #2 Failed

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

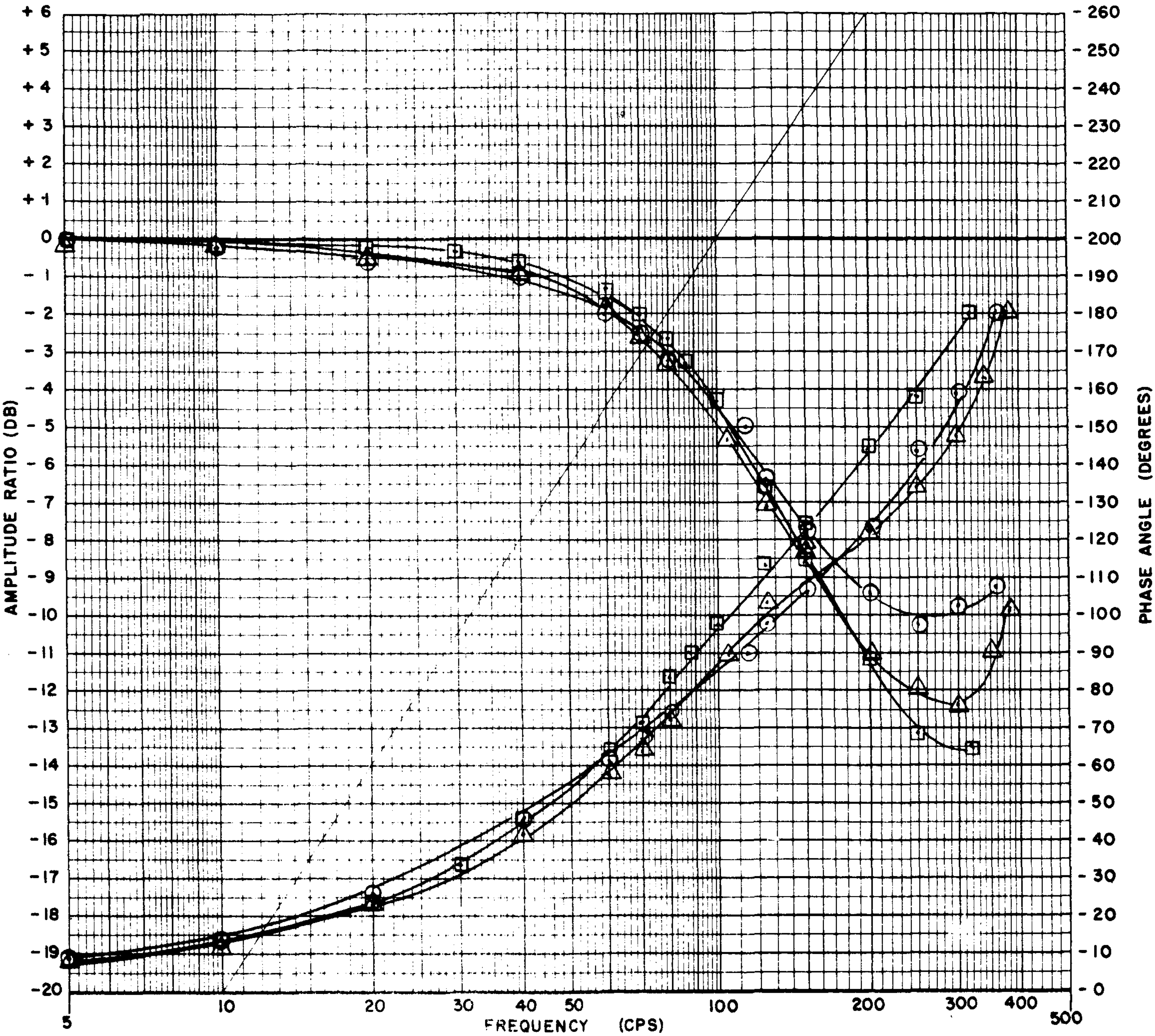
Fig. 15

○ - No Failure

△ - Open

□ - +50 ma

SERVO VALVE DYNAMIC RESPONSE



MODEL 16-147 SERIAL 3

DATE 6-2-65 BY lh

Torque Motor #3 Failed

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 1 MA.(P-P)

OUTPUT No Load Flow

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

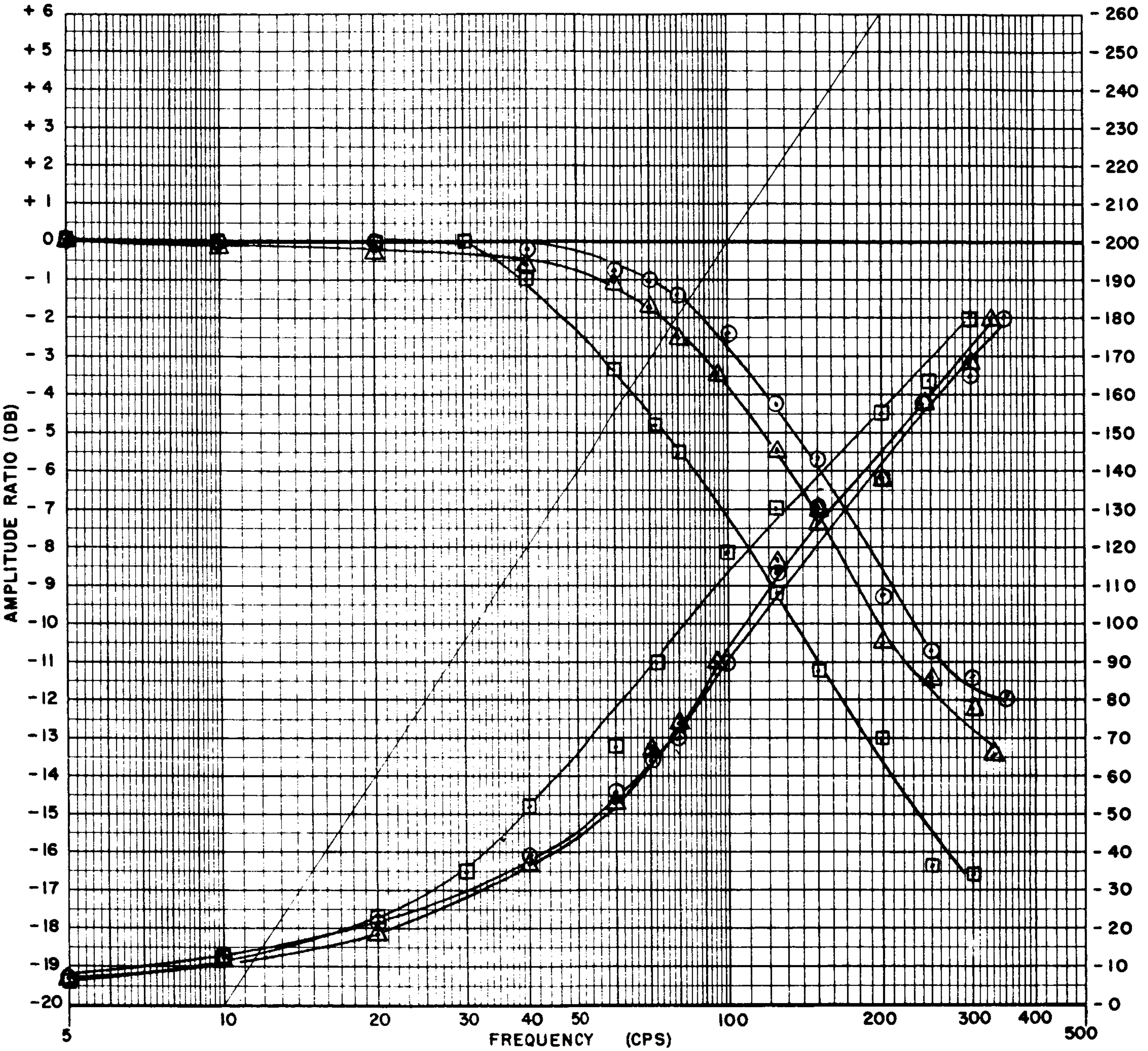
Fig. 16

○ - No Failure

△ - Open

□ - +50 ma

SERVO VALVE DYNAMIC RESPONSE



MODEL 16-147 SERIAL 3

DATE 6-2-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 10 MA.(P-P)

OUTPUT No Load Flow

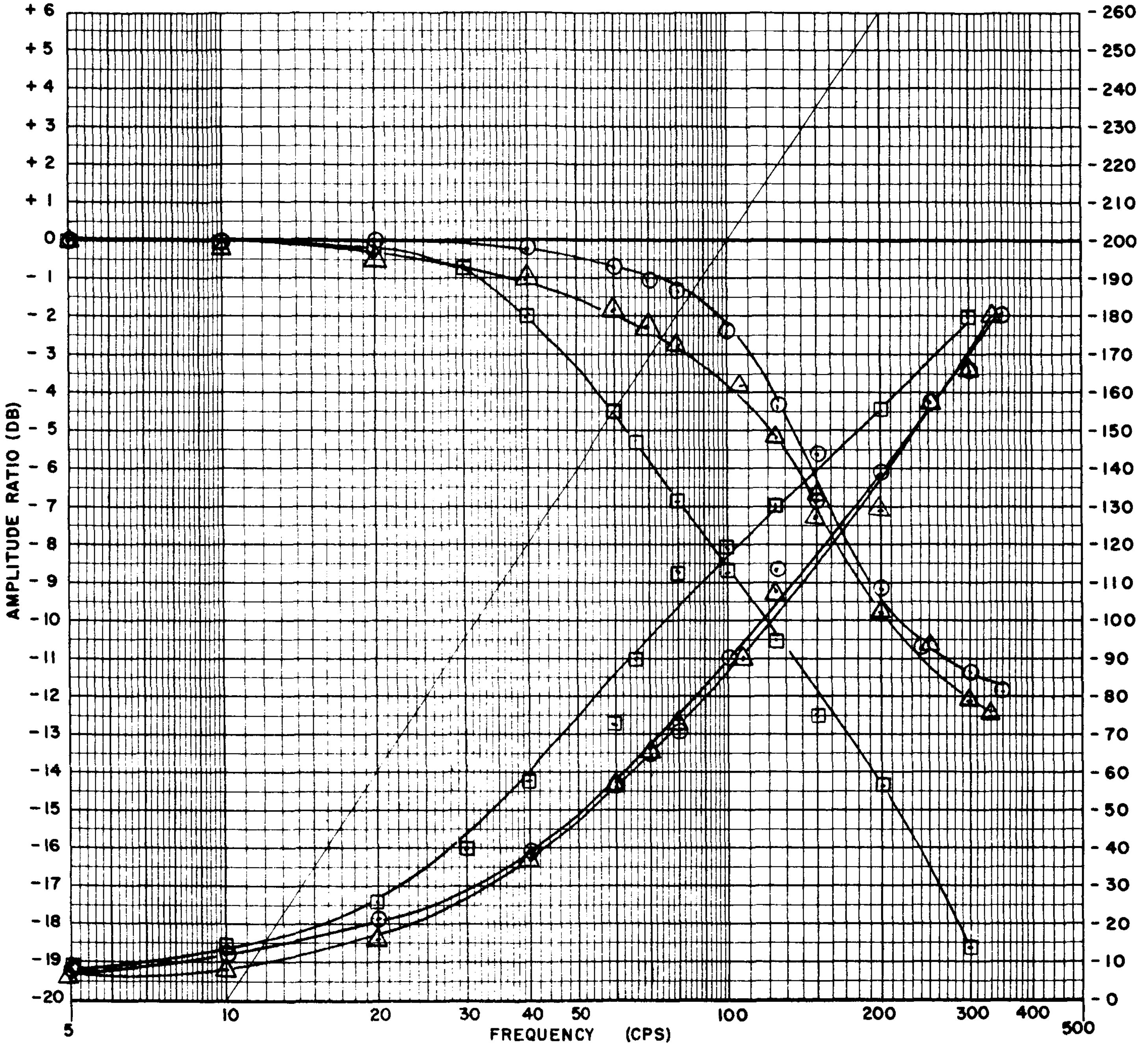
Torque Motor #1 Failed

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 17

○ - No Failure
△ - Open
□ - +50 ma

SERVO VALVE DYNAMIC RESPONSE



MODEL 16-147 SERIAL 3

DATE 6-2-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 10 MA.(P-P)

OUTPUT No Load Flow

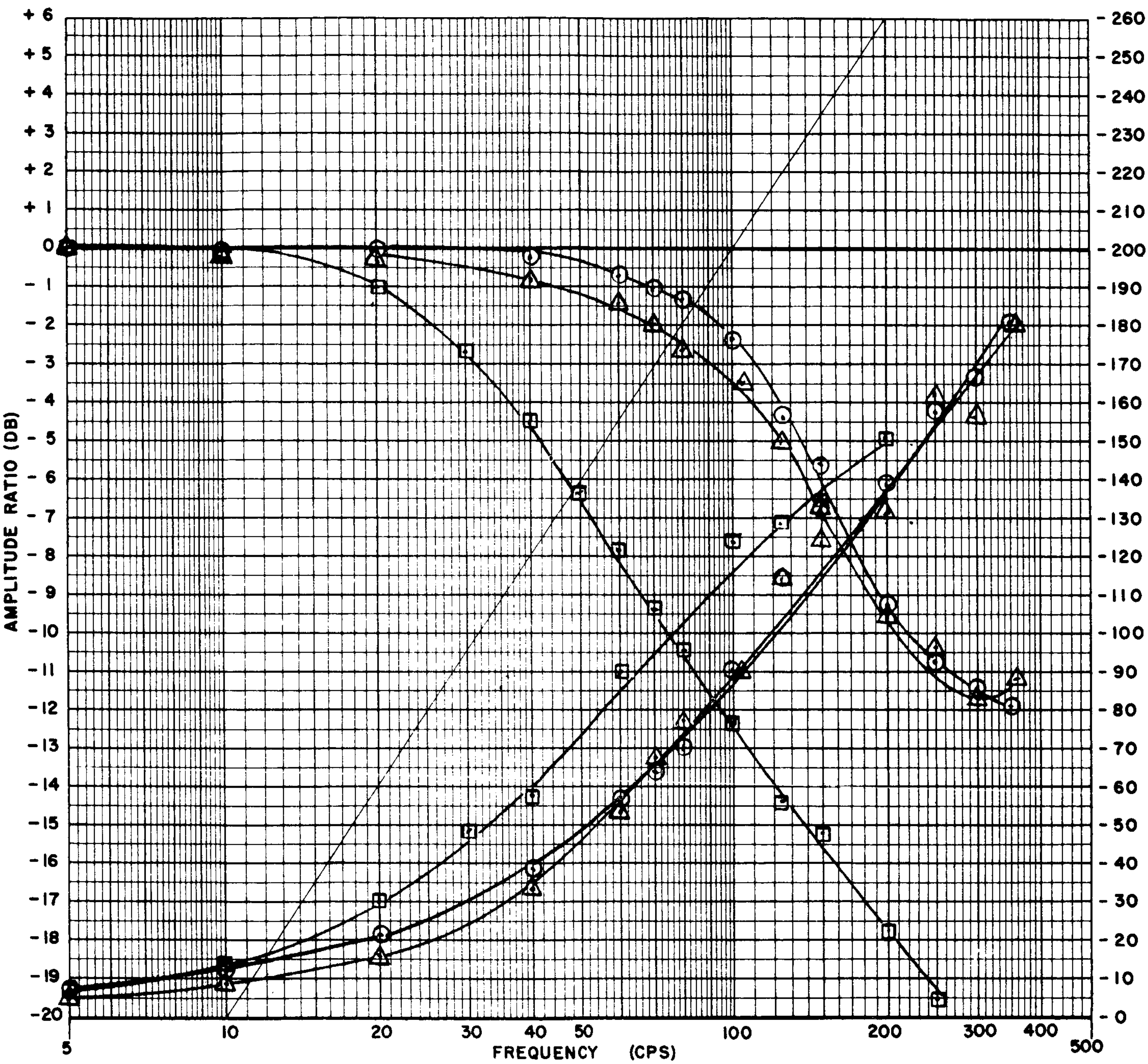
Torque Motor #2 Failed

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 18

○ - No Failure
△ - Open
□ - +50 ma

SERVO VALVE DYNAMIC RESPONSE



MODEL 16-147 SERIAL 3

DATE 6-2-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 10 MA.(P-P)

OUTPUT No Load Flow

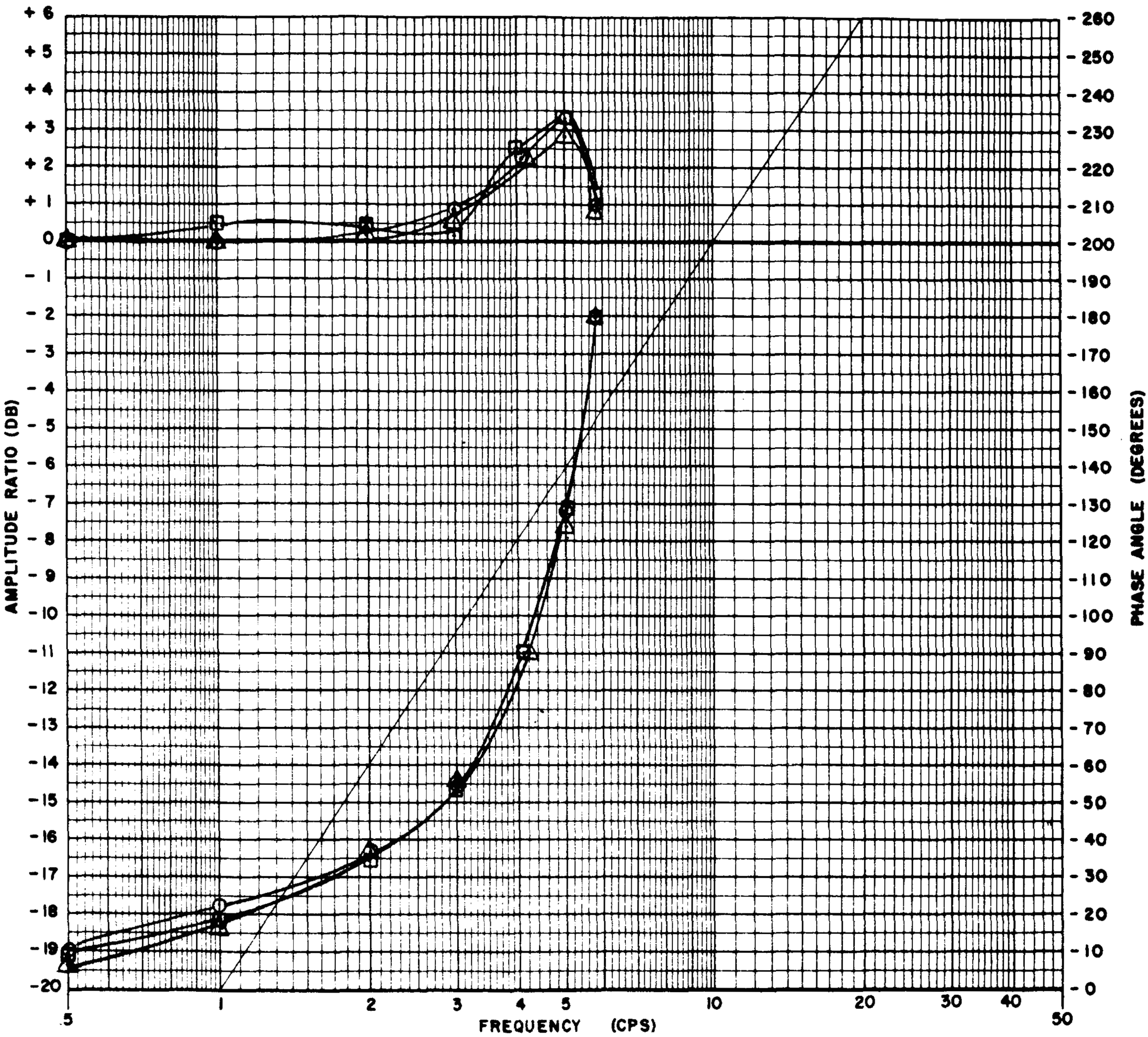
Torque Motor #3 Failed

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

Fig. 19

O - No Failure
△ - Open
□ - +50 ma

SYSTEM RESPONSE



○ - All Motors Active
K_{vx} = 15.8 sec^{-1}

△ - (+) Hard Over
□ - Open
} Torque Motor #1 Failed

MODEL 17X204 SERIAL 2

DATE 6-17-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 3.5 MA.(P-P)

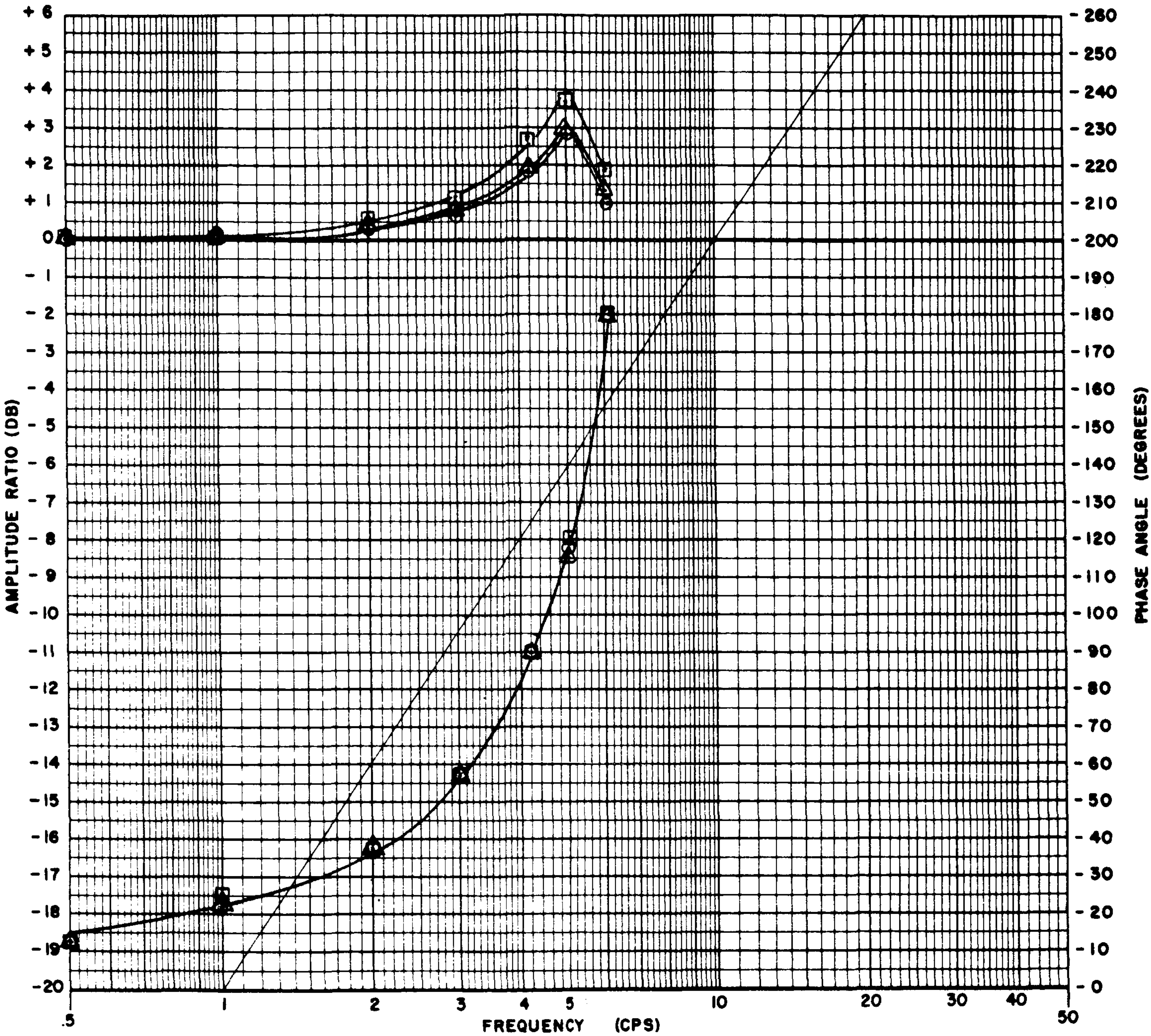
OUTPUT Simulator Load Position

Fig. 20

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

C

SYSTEM RESPONSE



○ - All Motors Active
K_{vx} = 15.8 sec^{-1}

△ - (+) Hard Over
□ - Open

} Torque Motor #1 Failed

MODEL 17X204 SERIAL 2

DATE 6-17-65 BY lh

SYSTEM PRESSURE 3500 PSI

INPUT SIGNAL 7.0 MA.(P-P)

OUTPUT Simulator Load Position

Fig. 21

MOOG SERVOCONTROLS, INC.
EAST AURORA, NEW YORK

C

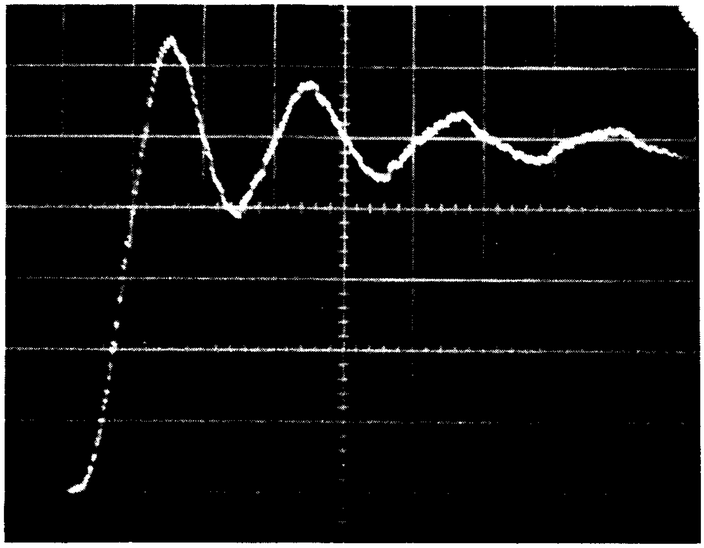


Fig. 22 Transient Response, All Channels Active
(3.5 ma p-p, time - 0.1 sec per large div.)

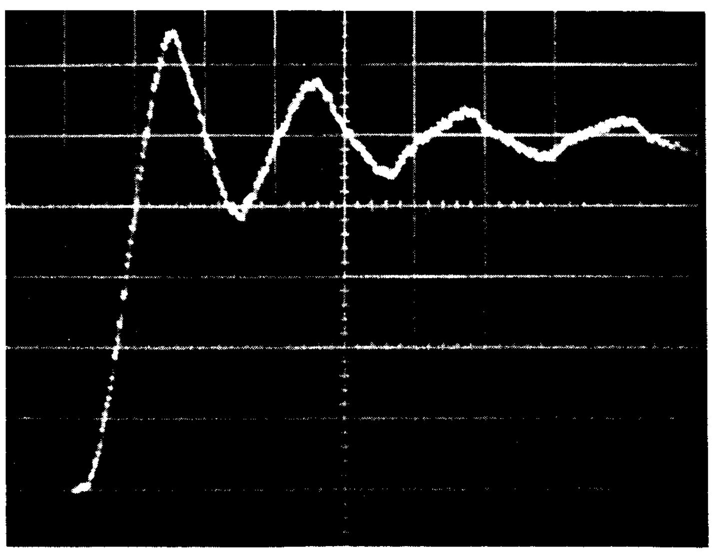


Fig. 23 Transient Response, Torque Motor #1 Failed Open
(3.5 ma p-p, time - 0.1 sec per large div.)

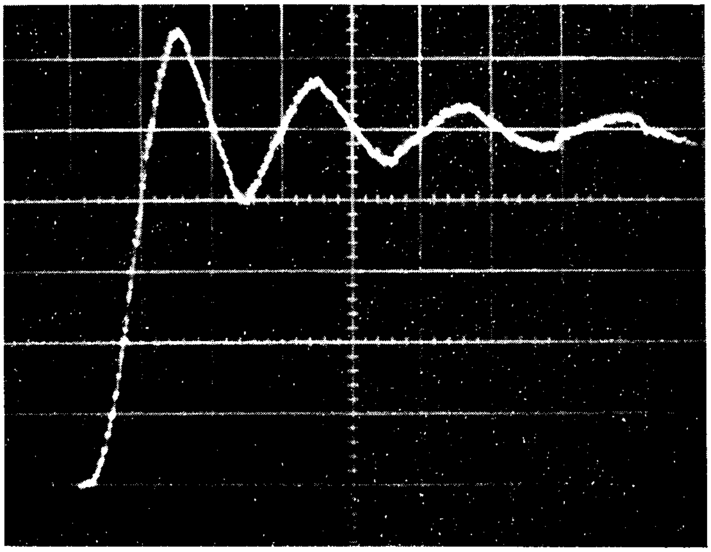


Fig. 24 Transient Response, Torque Motor #1 Failed Hard-Over (3.5 ma p-p, time - 0.1 sec per large div.)

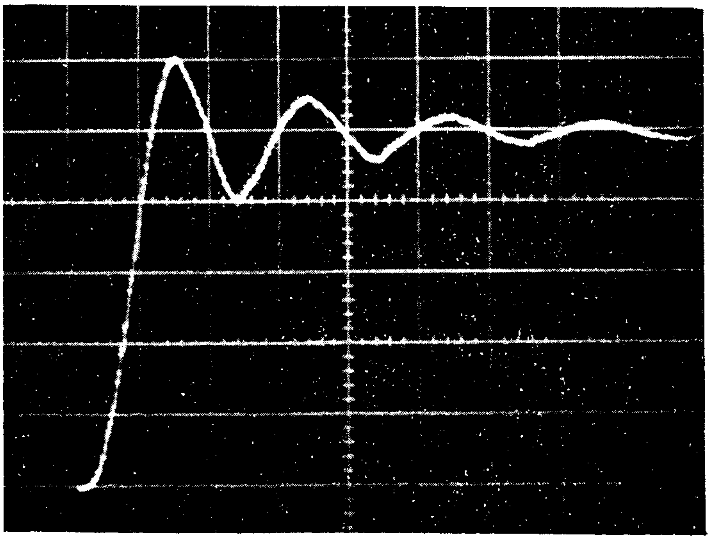


Fig. 25 Transient Response, All Channels Active (7.0 ma p-p, time - 0.1 sec per large div.)

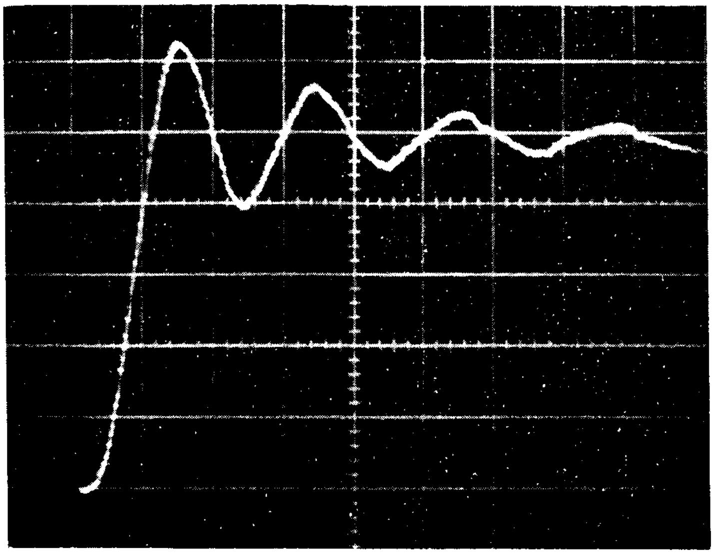


Fig. 26 Transient Response, Torque Motor #1 Failed Open
(7.0 ma p-p, time - 0.1 sec per large div.)

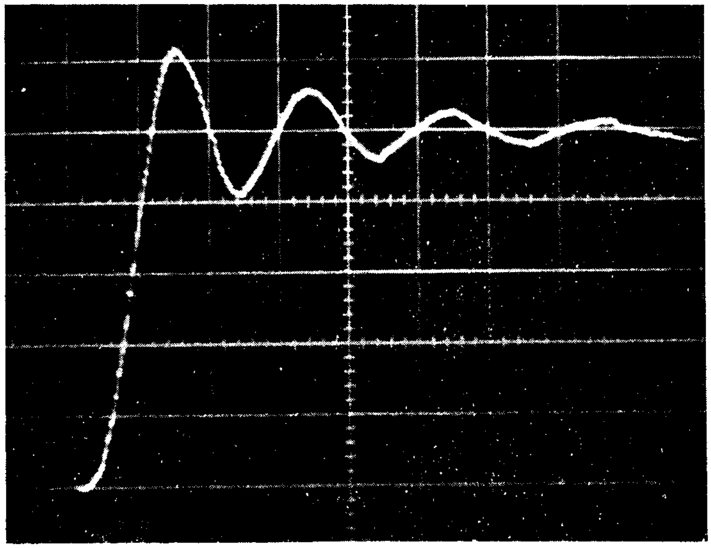


Fig. 27 Transient Response, Torque Motor #1 Failed Hard-Over
(7.0 ma p-p, time - 0.1 sec per large div.)

Temperature testing indicated less than 5% change in actuator position up to 275°F. Throughout the test program there was no evidence of abnormal or unstable performance of the servoactuators.

APPENDIX

I Performance Preductions and Sizing Calculations Report, ER-88.

II Revised Performance Preductions, ER-88A.

ENGINEERING REPORT NO. ER-88

Moog Servocontrols, Inc. East Aurora, New York

TITLE:

Performance Predictions and Sizing Calculations for Moog Model 17-204 Actuator; Mechanical Feedback Servoactuator with Majority Voting Valve.

BY: D. Gusakov AND B. H. Weyyon

DATE: October 9, 1964

REFERENCES:

(1) Proposal for Moog Model 17-204, Majority Voting Mechanical Feedback Servoactuator for the Saturn S-IV B. (Dated 5-5-64)

(2) Moog Servocontrols, Inc. Engineering Report No. ER-67, "Performance Predictions and Sizing Calculations for Saturn S-IV B TVC System: 17-189 Actuator, 16-142 Valve," dated August 22, 1963

(3) Moog Servocontrols, Inc. Engineering Report No. ER-67A, "Revision of Engineering Report ER-67 to reflect specification changes resulting from S-IV B Design Review Meeting on September 11 and 12, 1963," dated September 24, 1963

(4) Moog Servocontrols, Inc. Engineering Report No. ER-67B, "Revision of Engineering Report ER-67A to reflect changes resulting from revised S-IV B actuator specification, Revision F," dated April 2, 1964

SECTION 1.0: Summary

This report summarizes the performance predictions and sizing calculations for the Moog Servoactuator Model 17-204 to be supplied to the George C. Marshall Space Flight Center, National Aeronautics and Space Administration, Huntsville, Alabama. The design provides mechanical piston position feedback and a majority voting mechanical feedback servovalve. Where possible, the design has been patterned about the Saturn S-IV B Servoactuator and associated load characteristics (Moog Model 17-189 Servoactuator).

Section 2.0 contains a summary of the pertinent system parameters; and section 3.0 shows calculations of the various loop gains and sensitivities. Figure B illustrates the system function diagram of the unit.

SECTION 2.0: Summary of System Parameters

Symbol	Description	Value
A_1	DPF piston stub areas	$0.1295\ in^2$
A_2	DPF piston nozzle driving areas	$0.6560\ in^2$
A_{act}	actuator area	$11.78\ in^2$
D_N	DPF nozzle diameters	0.028 in
$k_{feedback\ spring - 1}$	Feedback spring stiffness Torque motor No. 1	29.3 lb/in.
$k_{feedback\ spring - 2}$	Feedback spring stiffness Torque motor No. 2	54.2 lb/in.
$k_{feedback\ spring - 3}$	Feedback spring stiffness Torque motor No. 3	29.3 lb/in.
k_{sfw-1}	Spring rate of spool feedback wire Torque motor No. 1	9.5 lb/in.
k_{sfw-2}	Spring rate of spool feedback wire Torque motor No. 2	9.5 lb/in.
k_{sfw-3}	Spring rate of spool feedback wire Torque motor No. 3	9.5 lb/in.
K_3	DPF pressure feedback slope	0.015 cis/psi
K_{cam}	feedback cam slope	0.04366 in/in. ($2° 30'$)
K_{DPF}	DPF torque/pressure gain	9.5×10^{-5} in-lb/psi
K_E	combined rate of DPF springs per unit	1000 lb/in.
K_M	actuator mechanical stiffness (calculated from specification parameters)	4,760,000 lb/in.
K_M'	estimated actual actuator mechanical stiffness	2,500,000 lb/in.

Symbol	Description	Value
K_{oil}	actuator oil stiffness	3, 390, 000 lb/in.
K_s	structural spring rate	391, 000 lb/in.
K_{spool}	valve spool gain - flow per displacement	2, 180 in^3/sec/in.
K_T	total drive stiffness (calculated from specification parameters)	350, 000 lb/in.
K_T'	estimated total drive stiffness	307, 000 lb/in.
K_{TM-1}	gain of torque motor No. 1	0. 046 in-lb/ma
K_{TM-2}	gain of torque motor No. 2	0. 046 in-lb/ma
K_{TM-3}	gain of torque motor No. 3	0. 046 in-lb/ma
K_{valve}	valve flow per torque gain	158 cis/in-lb.
$K_{vp}(DPF)$	dynamic pressure loop gain	33. 2 sec^{-1}
K_{vx}	positional loop gain	20 sec^{-1}
ℓ	total actuator stroke	2. 5 in.
$\ell_{feedback\ spring-1}$	length from feedback spring input to torque motor flexure point Torque motor No. 1	1. 16 in.
$\ell_{feedback\ spring-2}$	length from feedback spring input to torque motor flexure point Torque motor No. 2	0. 625 in.
$\ell_{feedback\ spring-3}$	length from feedback spring input to torque motor flexure point Torque motor No. 3	1. 16 in.
ℓ_{sfw-1}	length from spool displacement pickup point to torque flexure point Torque motor No. 1	1. 45 in.
ℓ_{sfw-2}	length from spool displacement pickup point to torque flexure point Torque motor No. 2	1. 45 in.

Symbol	Description	Value
ℓ_{sfw-3}	length from spool displacement pickup point to torque flexure point Torque motor No. 3	1.45 in.
M_E	equivalent reflected engine mass	120 $lb\text{-}sec^2/in.$
N	volumetric efficiency factor (assumed)	0.9 (dimensionless)
X_{DPF}	DPF nozzle spacing from flapper	0.0012 in.
β	bulk modulus of MIL-H-5606 oil	200,000 lb/in^2
τ_o	DPF time constant	0.08 sec. at ±200 psi load ΔP
ω_E	natural frequency of mass and effective oil-structure spring (specification value)	54 rad/sec.
ω'_E	natural frequency of mass and effective oil-structure spring (calculated from estimated parameters)	53.4 rad/sec.
ω_s	natural frequency of mass and structure spring	57.1 rad/sec.
	actuator sensitivity	0.031 in/ma
	rated command current	50 ma
	rated valve flow	10.5 gpm at 3500 psi Δ valve
	stroke of DPF piston	±0.26 in.
	actuator attach radius	11.87 inches
	engine gimballed inertia	1450 $slug\text{-}ft^2$
	valve spool diameter	0.375 inch

SECTION 3.0: Loop Gain and Sensitivities

Each of the three mass balanced -2 series torque motors has a nominal rated output of:

$$\text{Max. output torque} = (K_{TM})\,(\text{rated current})$$

$$= (0.046\ \text{in-lb/ma})\,(50\ \text{ma})$$

$$= 2.3\ \text{in-lb.}$$

The actuator ram position sensitivity to current input is nominally the same for each channel. While the actuator stroke is less than the S-IV B 17-189 unit, the sensitivity has been designed to be identical. Thus the piston will bottom before rated current. For channel -1:

$$\text{Actuator Sensitivity} = \frac{(K_{TM-1})}{(K_{cam})\,(k_{\substack{\text{feedback}\\ \text{spring - 1}}})\,(\ell_{\substack{\text{feedback}\\ \text{spring - 1}}})}$$

$$= \frac{0.046}{(0.04366)\,(29.3)\,(1.16)}$$

$$= 0.031\ \text{in/ma}$$

The position loop gain (K_{vx}) is nominally the same for each channel. For channel -1:

$$K_{vx} = \frac{K_{spool}\, K_{cam}\, k_{\substack{\text{feedback}\\ \text{spring - 1}}}\, \ell_{\substack{\text{feedback}\\ \text{spring - 1}}}}{A_{act}\, \ell_{sfw - 1}\, k_{sfw - 1}}$$

$$= \frac{(2180)\,(0.04366)\,(29.3)\,(1.16)}{(11.78)\,(1.45)\,(9.5)}$$

$$= 20\ \text{sec}^{-1}$$

The dynamic pressure loop gain K_{vp}(DPF) is nominally the same for each channel. For channel -1:

$$K_{vp}(DPF) = \frac{K_{spool}\, K'_T\, K_{DPF}}{A_{act}{}^2\, \ell_{sfw-1}\, k_{sfw-1}}$$

$$= \frac{(2180)\,(307,000)\,(9.5 \times 10^{-5})}{(11.78)^2\,(1.45)\,(9.5)}$$

The Moog servosystem analyzer was programmed to simulate the engine - stage structure characteristics of the Saturn S-IV B TVC and the 17-204 actuator, according to the nominal parameters as listed on Figure A. Step and frequency response data was taken and are presented in Figures A and AA respectively.

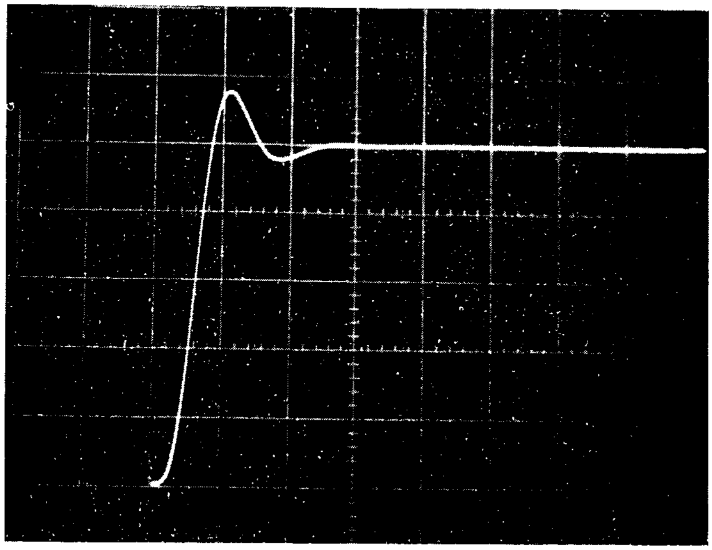


FIGURE A

Moog Model 17-204

Analog Computer Simulation of:

Engine Position Response to Step Command

All System Parameters Nominal as Follows:

Drive-Load Resonant Frequency = ω_e = 53.4 rad/sec.

Structure-Load Resonant Frequency = ω_s = 57.1 rad/sec.

Drive-Load Damping Ratio = ζ_e = 0

Positional Loop Gain = K_{vx} = 20 sec^{-1}

Dynamic Pressure Loop Gain = K_{vp}(DPF) = 33.2 sec^{-1}

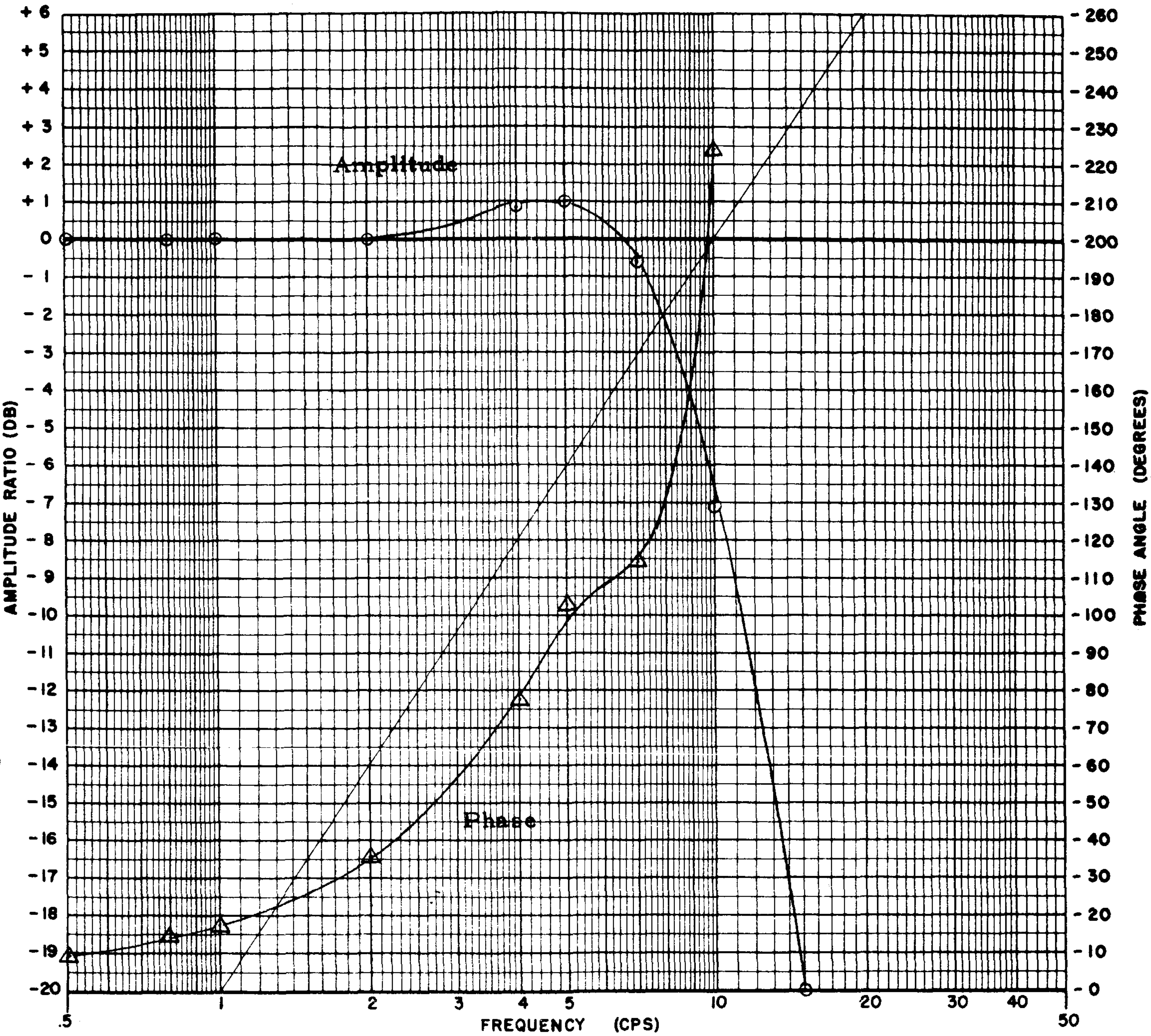
DPF Time Constant = τ_o = 0.08 sec at ±200 psi load Δ P

Servovalve dynamics: second order with a resonant frequency of 50 cps and a damping ratio of 1.0

Feedback from piston position

Single Sweep at 0.1 sec/cm

SERVOACTUATOR DYNAMIC RESPONSE



MODEL 17-204 SERIAL ________

DATE 8/26/64 BY IG/AH

SYSTEM PRESSURE ________ PSI

INPUT SIGNAL Command (βi) MA.(P-P)

OUTPUT Engine Position (β)

FIGURE AA
Analog Computer Simulation - Mechanical Feedback Servoactuator with Majority-Voting Valve.

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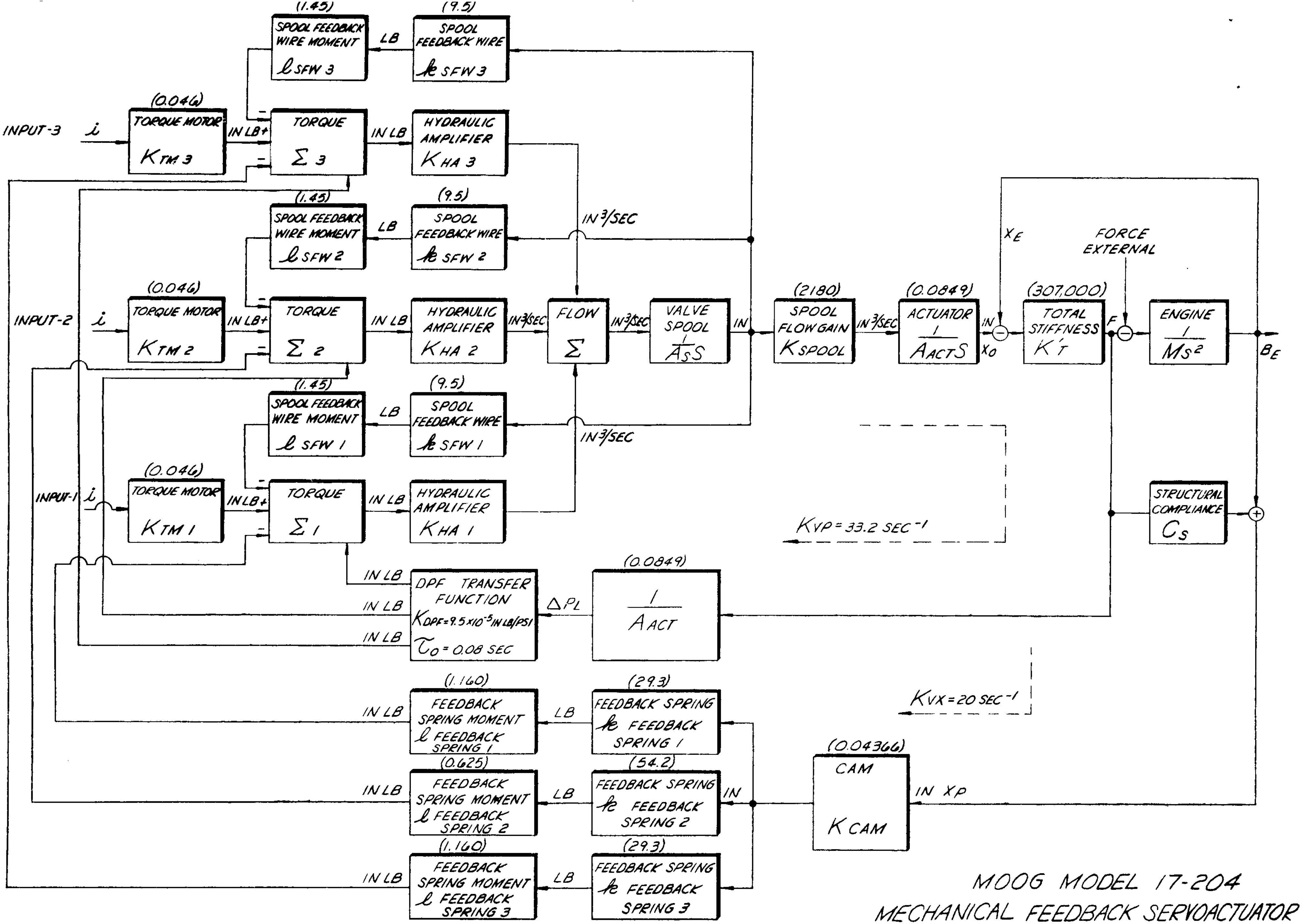
All parameters as listed on Figure A.



MOOG MODEL 17-204
MECHANICAL FEEDBACK SERVOACTUATOR
WITH MAJORITY VOTING VALUE

FIGURE-B SYSTEM FUNCTIONAL DIAGRAM

ENGINEERING REPORT NO. ER-88A

Moog Servocontrols, Inc. East Aurora, New York

TITLE:

Revision of Engineering Report ER-88 to Reflect Specification Changes of the Moog Model 17-204 Mechanical Feedback Actuator with Majority Voting Valve

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DATE: December 22, 1964

REFERENCE:

Moog Servocontrols, Inc. Engineering Report No. ER-88, "Performance Predictions and Sizing Calculations for Moog Model 17-204 Actuator; Mechanical Feedback Servoactuator With Majority Voting Valve"

SECTION 1.0 Introduction

This report summarizes the new performance predictions and system parameters of the Moog Model 17-204 servoactuator. The new performance predictions and system parameters reflect a lowering of the load mass and structure spring natural frequency (ω_s) from 57.1 rad/sec to 43.0 rad/sec and a lowering of the position loop gain (K_{vx}) from 20.0 sec^{-1} to 14.25 sec^{-1}.

Section 2.0 contains a summary of the pertinent system parameters. Section 3.0 shows the necessary calculations for determining values of the parameters which were affected by the above mentioned changes in ω_s and K_{vx}.

SECTION 2.0: Summary of System Parameters

Symbol	Description	Value
A_1	DPF piston stub areas	0.1295 in^2
A_2	DPF piston nozzle driving areas	0.6560 in^2
A_{act}	Actuator area	11.78 in^2
A_N	Valve nozzle areas	0.000201 in^2
D_N	DPF nozzle diameters	0.038 in
k_{B-1}	Effective Bernoulli spring rate Torque motor No. 1	1.72 lb/in
k_{B-2}	Effective Bernoulli spring rate Torque motor No. 2	1.72 lb/in
k_{B-3}	Effective Bernoulli spring rate Torque motor No. 3	1.72 lb/in
$k_{feedback\ spring-1}$	Feedback spring stiffness Torque motor No. 1	29.3 lb/in.
$k_{feedback\ spring-2}$	Feedback spring stiffness Torque motor No. 2	54.2 lb/in.
$k_{feedback\ spring-3}$	Feedback spring stiffness Torque motor No. 3	29.3 lb/in.
k_{sfw-1}	Spring rate of spool feedback wire Torque motor No. 1	12.4 lb/in
k_{sfw-2}	Spring rate of spool feedback wire Torque motor No. 2	12.4 lb/in
k_{sfw-3}	Spring rate of spool feedback wire Torque motor No. 3	12.4 lb/in
K_3	DPF pressure feedback slope	0.015 cis/psi
K_{cam}	Feedback cam slope	0.04366 in/in (2° 30')

Symbol	Description	Value
K_{DPF}	DPF torque/pressure gain	13.3×10^{-5} in-lb/psi
k_E	Combined rate of DPF springs per unit	479 lb/in
k_M	Estimated actual actuator mechanical stiffness	2,500,000 lb/in
k_{oil}	Effective actuator oil stiffness	3,390,000 lb/in
k_s	Effective structural spring rate	222,000 lb/in
K_{spool}	Valve spool gain - flow per displacement	2,180 in^3/sec/in
k_T	Estimated total drive stiffness	192,000 lb/in
K_{TM-1}	Gain of torque motor No. 1	0.046 in-lb/ma
K_{TM-2}	Gain of torque motor No. 2	0.046 in-lb/ma
K_{TM-3}	Gain of torque motor No. 3	0.046 in-lb/ma
K_{valve}	Valve flow per torque gain	113 cis/in-lb
$K_{vp}(DPF)$	Dynamic pressure loop gain	20.4 sec^{-1}
K_{vx}	Positional loop gain	14.25 sec^{-1}
ℓ	Total actuator stroke	2.5 in
ℓ_{f-1}	Length from flexure point to valve nozzles Torque motor No. 1	0.760 in
ℓ_{f-2}	Length from flexure point to valve nozzles Torque motor No. 2	0.760 in
ℓ_{f-3}	Length from flexure point to valve nozzles Torque motor No. 3	0.760 in
$\ell_{feedback\ spring-1}$	Length from feedback spring input to torque motor flexure point Torque motor No. 1	1.16 in

Symbol	Description	Value
$\ell_{\text{feedback spring-2}}$	Length from feedback spring input to torque motor flexure point Torque motor No. 2	0.625 in.
$\ell_{\text{feedback spring-3}}$	Length from feedback spring input to torque motor flexure point Torque motor No. 3	1.16 in
ℓ_{N-1}	Length from flexure point to DPF nozzles Torque motor No. 1	0.580 in
ℓ_{N-2}	Length from flexure point to DPF nozzles Torque motor No. 2	0.580 in
ℓ_{N-3}	Length from flexure point to DPF nozzles Torque motor No. 3	0.580 in
ℓ_{sfw-1}	Length from spool displacement pickup point to torque flexure point Torque motor No. 1	1.45 in.
ℓ_{sfw-2}	Length from spool displacement pickup point to torque flexure point Torque motor No. 2	1.45 in.
ℓ_{sfw-3}	Length from spool displacement pickup point to torque flexure point Torque motor No. 3	1.45 in.
M_E	Equivalent reflected engine mass	120 lb-sec^2/in.
N	Volumetric efficiency factor (assumed)	0.9 (dimensionless)
X_{DPF}	DPF nozzle spacing from flapper	0.0012 in.
β	Bulk modulus of MIL-H-5606 oil	200,000 lb/in^2
τ_o	DPF time constant	0.132 sec at ±200 psi load ΔP
ω_E	Natural frequency of mass and effective oil-structure spring (calculated from estimated parameters)	40.0 rad/sec.

Symbol	Description	Value
ω_s	Natural frequency of mass and structure spring	43 rad/sec.
	Actuator sensitivity	0.031 in/ma
	Rated command current	50 ma
	Rated valve flow	10.5 gpm at 3500 psi Δ valve
	Stroke of DPF piston	±0.26 in.
	Actuator attach radius	11.87 inches
	Engine gimballed inertia	1450 slug-ft^2
	Valve spool diameter	0.375 inch

SECTION 3.0: Calculation of System Parameters

Section 3.1: Determination of the natural frequency of mass and effective oil - structure spring (ω_E).

$$k_s = M_E \, \omega_s^2$$

$$k_s = (120)\,(43^2)$$

$$k_s = 222{,}000 \text{ lb/in.}$$

$$k_{oil} = 3{,}390{,}000 \text{ lb/in.}$$

$$k_M = 2{,}500{,}000 \text{ lb/in}$$

$$k_T = \frac{1}{\frac{1}{k_M} + \frac{1}{k_{oil}} + \frac{1}{k_s}}$$

$$k_T = \frac{1}{\frac{1}{2{,}500{,}000} + \frac{1}{3{,}390{,}000} + \frac{1}{222{,}000}}$$

$$k_T = 192{,}000 \text{ lb/in}$$

$$\omega_E = \sqrt{k_T/M_E}$$

$$\omega_E = \sqrt{\frac{192{,}000}{120}}$$

$$\omega_E = 40.0 \text{ rad/sec}$$

Section 3.2: Determination of the effective Bernoulli spring rate within the servovalve.

The Bernoulli force on the valve spool F_B can be expressed by the relationship:

$$F_B = 0.45\, W\, (P_s - P_L)\, X_s$$

where:

0.45 is a constant
$W = 0.6$ in. is the total slot width
$P_s = 3500$ psi is the system pressure
$P_L = 0$ is the load pressure drop
X_s is the spool position.

The torque on the armature, T_B, due to the change in pressure across the nozzles caused by the Bernoulli force on the spool is:

$$T_B = \frac{F_B \, A_N \, \ell_f}{A_s}$$

where:

$A_N = 0.000201 \text{ in}^2$ is the nozzle area
$A_s = 0.1104 \text{ in}^2$ is the spool area
$\ell_f = 0.76$ in. is the distance from the flexure point to the nozzles or the Bernoulli moment

Assuming that $P_L = 0$,

$$T_B = \frac{0.45 \, W \, P_s \, X_s \, A_N \, \ell_f}{A_s}$$

$$\frac{T_B}{X_s} = \frac{(0.45)(0.6)(3500)(0.000201)(0.76)}{0.1104}$$

$$\frac{T_B}{X_s} = 1.307 \ \frac{\text{in-lb}}{\text{in}}$$

$$k_B = \frac{F_B}{X_s}$$

$$k_B = \frac{T_B}{X_s \, \ell_f}$$

$$k_B = \frac{1.307}{0.76}$$

$k_B = 1.72$ lb/in

The effect of this spring rate adds to that of the spool feedback wire rate of the valve. This is shown on the block diagram of Figure B. k_B and ℓ_f have the same values for each of the three servovalves.

Section 3.3: Determination of the spool feedback wire rate, k_{sfw}

The position loop gain K_{vx} is defined by:

$$K_{vx} = \frac{K_{spool} K_{cam} k_{fbs} \ell_{fbs}}{A_{act} (k_{sfw} \ell_{sfw} + k_B \ell_f)}$$

$$k_{sfw} = \frac{K_{spool} K_{cam} k_{fbs} \ell_{fbs}}{K_{vx} A_{act} \ell_{sfw}} - \frac{k_B \ell_f}{\ell_{sfw}}$$

$$k_{sfw} = \frac{(2180)(0.04366)(29.3)(1.16)}{(14.25)(11.78)(1.45)} - \frac{(1.72)(0.76)}{1.45}$$

$$k_{sfw} = 13.31 - 0.90$$

$$k_{sfw} = 12.4 \text{ lb/in.}$$

k_{sfw} has the same value for all three servovalves.

Section 3.4: Determination of the dynamic pressure loop gain K_{vp}DPF.

$$K_{vp} = \frac{K_3 k_T}{A_{act}^2}$$

$$K_{vp} = \frac{(0.015)(192,000)}{11.78^2}$$

$K_{vp} = 20.4 \text{ sec}^{-1}$ for each of the three loops

Section 3.5: Determination of the valve flow per torque gain, K_{valve}.

$$K_{valve} = \frac{K_{spool}}{k_{sfw} \ \ell_{sfw} + k_B \ \ell_f}$$

$$K_{valve} = \frac{2180}{(12.4)(1.45) + (1.72)(0.76)}$$

$K_{valve} = 113$ cis/in-lb for each of the three servovalves.

Section 3.6: Determination of the DPF torque per pressure gain, K_{DPF}.

$$K_{DPF} = \frac{K_3 (k_{sfw} \ \ell_{sfw} + k_B \ \ell_f)}{K_{spool}}$$

$$K_{DPF} = \frac{(0.015) \ [(12.4)(1.45) + (1.72)(0.76)]}{2180}$$

$K_{DPF} = 13.3 \times 10^{-5}$ in-lb/psi

Section 3.7: Determination of the DPF nozzle diameter, D_N.

$$D_N = \sqrt{\frac{4 K_3 A_2}{\pi \ \ell_N K_{valve} A_1}}$$

$$D_N = \sqrt{\frac{(4)(0.015)(0.656)}{\pi (0.580)(113)(0.1295)}}$$

$D_N = 0.038$ in

Section 3.8: Determination of the combined rate of the DPF springs, k_E.

From Figure C, the nominal DPF time constant is $\tau_o = 0.132$ sec.

$$k_E = \frac{2A_2^{\,2}\sqrt{\left(\frac{A_1}{A_2}\right)\Delta P_{LO}}}{(3)\,(209)\,\pi\, D_N\, X_{DPF}\, \tau_o} \quad \text{where: } \Delta P_{LO} = 200 \text{ psi}$$

$$k_E = \frac{(2)\,(0.656)^2\sqrt{\left(\frac{0.1295}{0.656}\right)(200)}}{(3)\,(209)\,\pi\,(0.038)\,(0.0012)\,(0.132)}$$

$$k_E = 479 \text{ lb/in}$$

Section 3.9: Determination of the static pressure which will cause bottoming of the DPF piston.

$$\Delta P = \frac{k_E\, X_p}{A_1}$$

$$= \frac{(479 \text{ lb/in})\,(\pm 0.26 \text{ in})}{(0.1295)}$$

$$= \pm\, 960 \text{ psi}$$

The Moog servosystem analyzer was programmed to simulate the engine-stage structure characteristics of the Saturn S-IVB TVC and the 17-204 actuator according to the current nominal parameters which are listed in Figure A. Step and frequency response data were taken and are presented in Figures A and AA respectively.

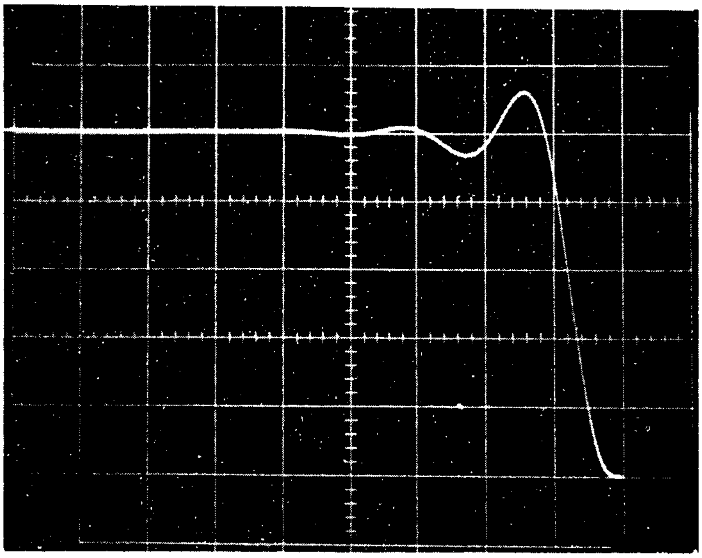

FIGURE A

Moog Model 17-204 Servoactuator

Analog Computer Simulation of:

Engine Position Response to Step Command

All System Parameters Nominal as Follows:

Drive - Load Resonant Frequency $= \omega_E = 40.0$ rad/sec

Structure - Load Resonant Frequency $= \omega_s = 43.0$ rad/sec

Drive - Load Damping Ratio $= \zeta_e = 0$

Position Loop Gain $= K_{vx} = 14.25\ sec^{-1}$

Dynamic Pressure Loop Gain $= K_{vp}$ (DPF) $= 20.4\ sec^{-1}$

DPF Time Constant $= \tau_o = 0.132$ sec at ± 200 psi load ΔP

Servovalve Dynamics : Second Order with a Resonant Frequency of 50 cps and a damping ratio of 1.0

Feedback From Piston Position

Single Sweep at 0.1 sec/cm

SERVOACTUATOR DYNAMIC RESPONSE

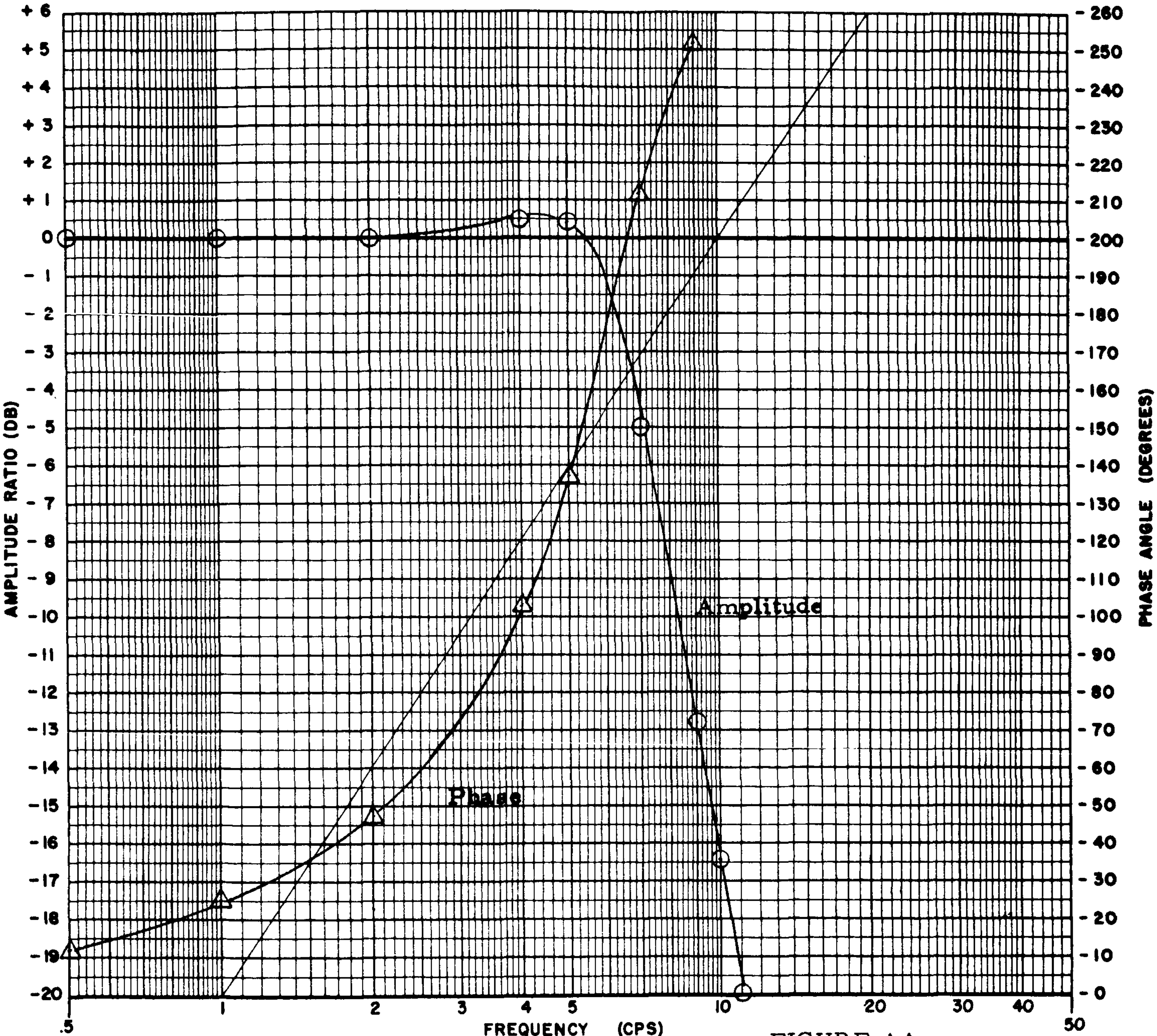

FIGURE AA

Analog Computer Simulation - Mechanical Feedback Servoactuator With Majority - Voting Valve.

MODEL 17-204 SERIAL ____

DATE 12/19/64 BY ____

SYSTEM PRESSURE ____ PSI

INPUT SIGNAL Command (θ_i) MA.(P-P)

OUTPUT Engine Position (θ_e)

MOOG SERVOCONTROLS, INC.
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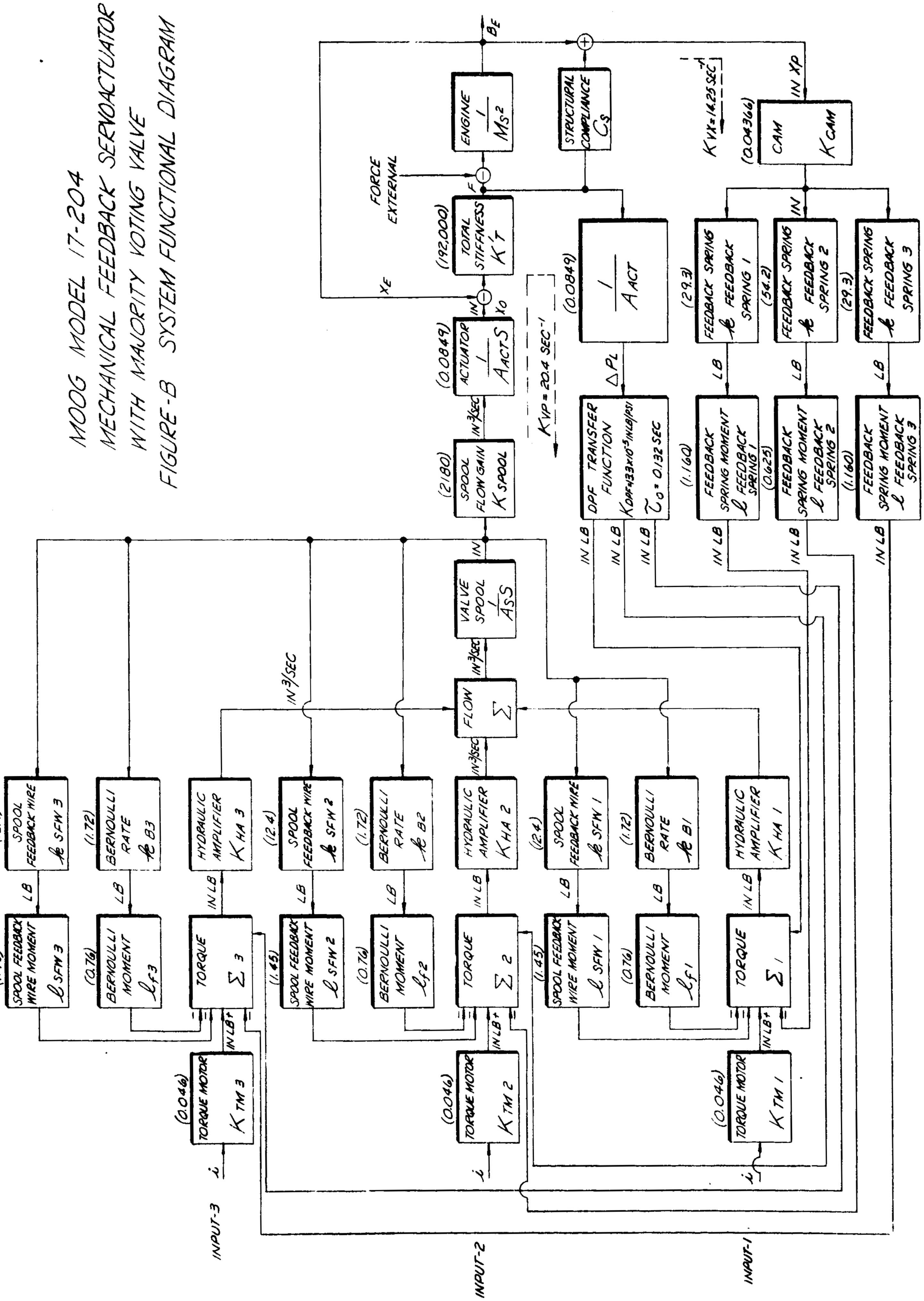
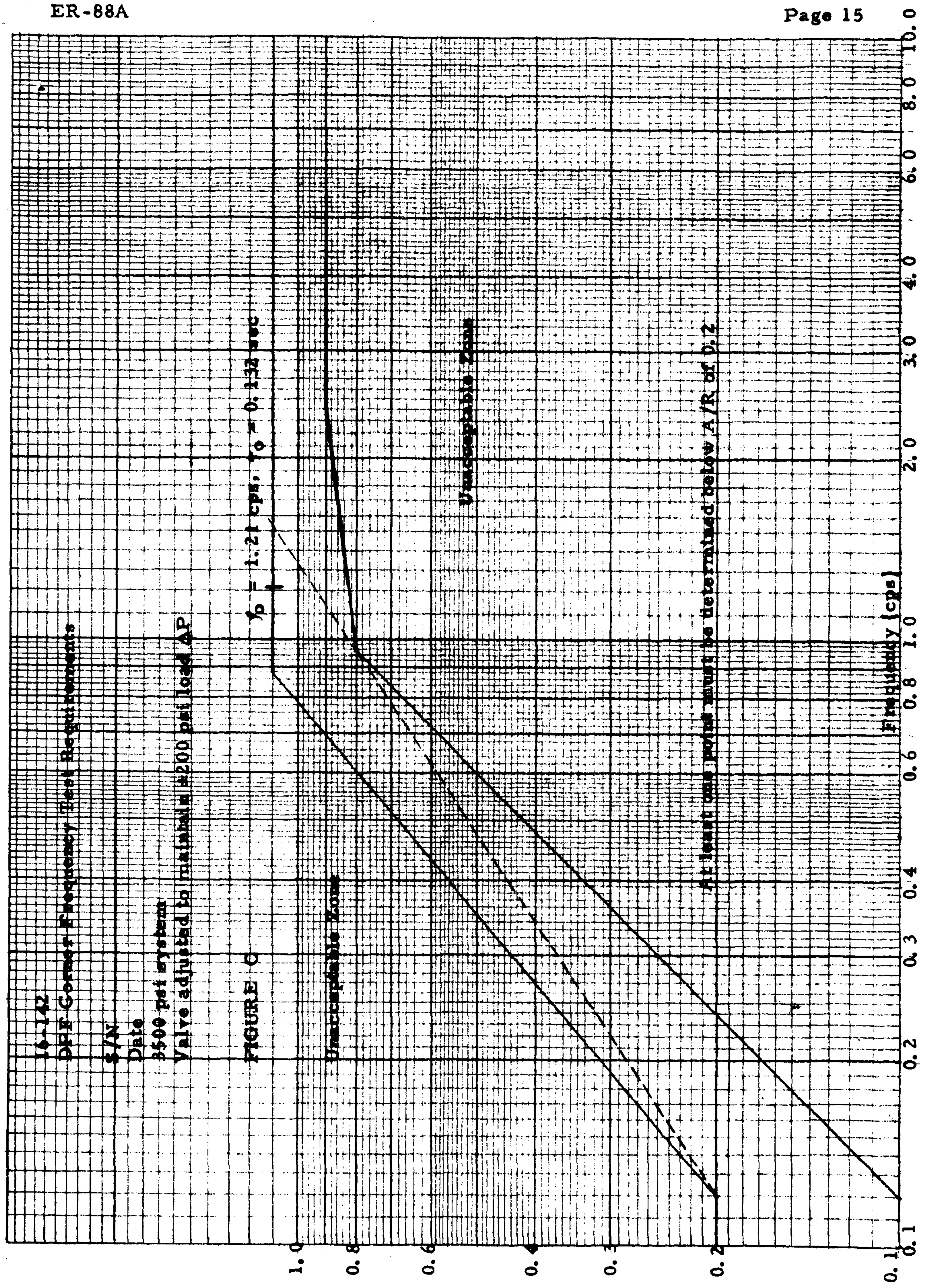
All parameters as listed on Figure A



MOOG MODEL 17-204
MECHANICAL FEEDBACK SERVOACTUATOR
WITH MAJORITY VOTING VALVE
FIGURE-B SYSTEM FUNCTIONAL DIAGRAM


16-142
DFF Corner Frequency Test Requirements
S/N
Date
3500 psi system
Valve adjusted to maintain ±200 psi load ΔP
FIGURE C
Unacceptable Zone
f_b = 1.21 cps, τ_b = 0.132 sec
Unacceptable Zone
At least one point must be determined below A/R of 0.2
Frequency (cps)
0.1
0.2
0.3
0.4
0.6
0.8
1.0
2.0
3.0
4.0
6.0
8.0
10.0
0.1
0.2
0.3
0.4
0.6
0.8
1.0